GAS TURBINE CONSTRUCTION

Including Operation and Maintenance

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New York : 1947

PRENTICE-HALL, INC.

PRINTED IN THE UNITED STATES OF AMERICA

TO MOTHER

Foreword

This book is a valuable adjunct to the author's *The Modern Gas Turbine.* It is indeed almost a necessity to the many who today are following with unparalleled interest the activity in gas turbine development. *The Modern Gas Turbine* was published only about two and a half years ago. There was an immediate and gratifying response. The engineer and the layman alike appreciated the opportunity to visualize the even then apparent oncoming trend in the practical commercial development and utilization of this hitherto sleeping giant of industry. When Sir Harold Clapp wrote in the foreword of *The Modern Gas Turbine:* "In a very few years we may see it (the gas turbine) driving our locomotives, ships, and planes," the statement was regarded by many as wishful thinking.

Today gas turbines are under construction by several American manufacturers to fulfill bonafide orders for all three of these transportation requirements. This actuality of progress in construction and performance is what makes this book a "must" for those desirous of keeping even partially abreast of this rapidly changing progressive picture. *Gas Turbine Construction* should not be considered a book solely for the construction or maintenance engineers to whom the subject might seem primarily to be addressed. As was *The Modern Gas Turbine,* it is authentic and factual and of interest to both engineer and layman.

Just one personal word. As Tom Sawyer's father I am naturally, and I think rightfully, proud of him and his many accomplishments. As I myself am cognizant of the practicable adaptability of the gas turbine as an efficient prime mover, I have followed his analysis and reasoning with great interest. Today, the successful gas turbine is a reality, no longer a

dream of the theorist, and I think that I can say fairly and without being considered parentally prejudiced that this book and *The Modern Gas Turbine* give all of us a far better understanding and prospective of today's realization and the future trend of this most modern development.

Willits H. Sawyer
Executive Engineer
Fellow, A.I.E.E. and A.S.M.E; E.D.

Acknowledgments

For assistance and cooperation in the preparation of this companion to *The Modern Gas Turbine,* I am indebted to practically everyone who aided me in writing that volume. Particularly do I wish to express my appreciation to my many associates, including J. B. Frauenfelder, R. F. Gagg, F. T. Hague, L. B. Jackson, G. F. McGowan, J. T. Rettaliata, D. M. Shackelford, and R. B. Smith.

Special thanks are due G. Geoffrey Smith, Directing Editor of *Flight* and author of *Gas Turbines and Jet Propulsion for Aircraft,* 4th edition, published in the United States by Aircraft Books, Inc., New York.

Specifically I am indebted to the following:

Automotive and Aviation Industries magazine; especially R. K. Winkleblack, for the table in Chapter I and illustrations in Chapter XIII.

Sun Oil Company; especially A. E. Pew, Jr., Vice President; H. E. Melton; and C. H. Thayer for the material contained in Chapter II.

Brown Boveri Corporation; especially Paul R. Sidler, President, for Chapters IV and V.

The Elliott Company; especially J. F. Cunningham, Jr.; and R. A. Riester for the information contained in Chapter VI, and C. F. Harms, for Chapter IX. Credit should also be given Lt. J. H. Gray, U. S. Navy, for his part in the preparation of Chapter VI.

Escher Wyss Corporation; especially Dr. C. Keller, for Chapter VII.

Wright Aeronautical Corporation; especially Paul Bancel, for the material contained in Chapter XI.

D. M. Shackelford, for Chapter XII.

Westinghouse Corporation; especially J. S. Newton, T. J.

Putz, and R. B. Rogers for the information in Chapters XIII and XIV.

Flight magazine; especially G. Geoffrey Smith, Directing Editor, for material and illustrations contained in Chapters XVI through XXII.

Others whose counsel, aid, and inspiration made this book possible are:

K. A. Browne, Chesapeake and Ohio Railway Company
William Dixey, General Electric Company
W. Giger, Allis-Chalmers Manufacturing Company
H. Kasschau, Consultant
L. N. Rowley, *Power* Magazine
J. K. Salisbury, General Electric Company.
J. W. Schaeffer, General Electric Company
S. A. Tucker, McGraw-Hill Publishing Company
W. B. Tucker, Allis-Chalmers Manufacturing Company

R. Tom Sawyer

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Chapter I

Introduction

The first part of this book describes heavy equipment which may be used in stationary, marine, or locomotive service. Very little difference exists in the equipment designed for these three classes of service. The differentiation is based on the design produced by the various manufacturers rather than on that demanded by the type of service.

For example, the Allis-Chalmers Company prefers an axial compressor for any class of heavy-duty service, yet it has built turbosuperchargers and jet units with a centrifugal compressor. The turbine may be of the single-stage type for jet propulsion and turbosupercharger or it may be of the multistage type. The multistage turbine may have all reaction blades or, as in the unit built for the United States Navy, Annapolis Testing Laboratory, there may be two rows of impulse and several rows of reaction blades. This unit was designed for a temperature of 1500° F, and it is desirable to drop this temperature as quickly as possible as it passes through the turbine, which is the reason for having the two rows of impulse blades.

The Elliott Company also uses the centrifugal compressor for its turbochargers, and it builds a very complete line primarily for heavy-duty Diesel engine supercharging. This company proposes to use the centrifugal compressor on locomotives or on any other unit where weight is of extreme importance, since the centrifugal compressor is much lighter than any other type, even though it may not be quite so efficient at certain load conditions.

For marine use, the Elliott Company is furnishing its positive-displacement compressor designed by Dr. Lysholm.

Unusually efficient for a positive-displacement type, this compressor has a very flat, stable curve for variable-speed operation. The Lysholm compressor, with or without the gas turbine drive, is ideally suited to certain industrial applications.

Dr. Curt Keller, of Escher Wyss, prefers the axial-flow compressor and multistage turbine for the closed-cycle system because in this system the units have to deal with unusually high pressures. The closed cycle has practically the same maintenance problems as the open cycle, the chief difference being in the main heater and controls. The closed cycle is well suited to certain types of marine and stationary service. Helium, for example, may be used in the closed cycle to cool a uranium pile that produces atomic energy.*

Turbochargers and turbosuperchargers are now widely used for supercharging all types of engines. It is now common to use the name *turbosupercharger* on all turbo units used to supercharge aviation engines which units deliver a high pressure of at least 15 lb to the engine intake. The name *turbocharger* is used for the turbo unit used to supercharge all other engines which unit delivers a normal pressure of 5 to 20 lb to the engine intake. Turbochargers have now been in extensive service on Diesel engines for over 10 years, and in most cases the original turbochargers are still in use, and are giving a good account of themselves. The Buchi type is the standard design for Diesel engines, and it can be stated that the turbocharged engine costs less to maintain per horsepower output than the same engine not turbocharged. Longevity and economy in operation indicate that a gas turbine power plant will cost less to maintain than a Diesel plant provided that the gas turbine plant is properly designed for the service in which it is placed.

Turbochargers in production are given a quick shop test, but it is necessary to make elaborate tests of turbochargers during development. Consequently, a chapter, written by one who spent several years making these laboratory tests, has

* For further discussion of the application of gas turbines to a uranium pile, see Smith, Fox, Sawyer, Austin, *Applied Atomic Power.* New York: Prentice-Hall, Inc., 1946.

TABLE 1

AIRCRAFT GAS TURBINE ENGINES

United States, England, Germany

			Performance at Sea-Level Static Conditions				Number of Stages		Combustion Chamber		Dimensions and Weights		
Country and Manufacturer	Model Number	Type of Propulsion Unit	Thrust (Lb)	Rpm	Specific Fuel Consumption (Lb per hr per lb of thrust)	Type of Com-pressor	Com-pressor	Turbine	Number	Type	Over-all Diameter (In.)	Over-all Length (In.)	Over-all Weight (Lb)
UNITED STATES General Electric Co.	I–16	J	1,650	16,500	1.18	Cent	DE	1	10	RF	41.50	70.0	825
General Electric Co.	I–40	J	4,000	11,500	1.185	Cent	DE	1	14	DF	48.00	101.5	1,820
General Electric Co.	TG–100	PJ	(2,200 hp) +600	13,000		Ax	11	1	8	RF	38.00	113.0	2,000
General Electric Co.	TG–180	J	4,000	7,600		Ax	11	1	8	DF			2,300
Westinghouse Elec. Mfg. Co.	9.5B	J	275	34,000	1.50	Ax	6	1	1	An	9.50	52.5	140
Westinghouse Elec. Mfg. Co.	19A	J	1,360	18,000	1.35	Ax	6	1	8	DF	19.00	100.0	830
Westinghouse Elec. Mfg. Co.	19B	J	1,365	18,000	1.28	Ax	6	1	1	An	19.00	89.8	810
GREAT BRITAIN Armstrong-Siddeley	ASX	J	2,600	8,000	1.03	Ax	14	2	11	RF	42.00	168.0	1,900
Armstrong-Siddeley	Python	PJ	(3,670 hp) +1,150	8,000		Ax	14	2	11	RF	48.00	136.0	3,950
Armstrong-Siddeley	Mamba	PJ	(1,000 hp) +320						6		27.00		750

(Table 1 continued on page 4)

TABLE 1

AIRCRAFT GAS TURBINE ENGINES (*Cont.*)

United States, England, Germany

Country and Manufacturer	Model Number	Type of Propulsion Unit	Performance at Sea-Level Static Conditions: Thrust (Lb)	Rpm	Specific Fuel Consumption (Lb per hr per lb of thrust)	Type of Compressor	Number of Stages: Compressor	Turbine	Combustion Chamber: Number	Type	Dimensions and Weights: Over-all Diameter (In.)	Over-all Length (In.)	Over-all Weight (Lb)
Bristol	Theseus 1	PJ	(1,950 hp) +500	8,200	0.57	Ax Cent	9 1	2	9	RF	49.00	106.0	2,310
Bristol	Proteus	PJ											
de Havilland	Goblin II	J	3,000	10,200	1.23	Cent	SE	1	16	DF	50.00	107.0	1,550
de Havilland	Ghost II	J	5,000		1.05	Cent	SE	1	10	DF	53.00	122.5	1,950
Metropolitan-Vickers	F 2/3	AJ	4,000	7,600	0.65	Ax	9	6	1	An	47.00	137.0	2,200
Metropolitan-Vickers	F 2 Ser. IV	J	3,500	7,400	1.05	Ax	10	1	1	An	37.11	159.0	1,750
Power Jets Ltd.	W 2/700	J	2,150	16,750	1.05	Cent	DE	1	10	RF	54.00	61.5	870
Rolls Royce	Welland 1	J	1,450	17,100		Cent	1	1	10	RF	43.00		850
Rolls Royce	Derwent 1	J	2,000	16,600		Cent	1	1	10		43.00		920
Rolls Royce	Derwent V	J	3,500	14,600	1.00	Cent	DE	1	9	DF	43.00	83.1	1,250
Rolls Royce	Nene	J	5,000	12,300	1.065	Cent	DE	1	9	DF	49.50	96.1	1,550
Rolls Royce	Trent 1	PJ	1,250			Cent	1	1	9				1,250
Rolls Royce	Clyde	PJ	(3,000 hp) +1,200	6,000		Ax Cent	9 1	2	9				2,500
Rolls Royce	Dart	PJ											

GERMANY													
BMW	003A2	J	1,760	9,500	1.45	Ax	7	1	1	An	27.10	124.0	1,344
BMW	003C	J	1,990	9,500	1.45	Ax	7	1	1	An	27.10	118.0	1,344
BMW	003D	J	2,260	10,000		Ax	8	2	1	An	27.50	125.0	1,366
BMW	003R	RJ	4,510			Ax	7	1	1	An			1,430
BMW	018	J	7,520	5,000		Ax	12	3	1	An	49.25	167.0	6.400
BMW	028	PJ	2,200	7,000		Ax	12	4	1	An			7,720
Daimler-Benz	007	AJ	1,760	12,000	1.35	Ax	9	1	4	DF	35.40	196.80	2,866
Heinkel-Hirth	011	J	2,860	10,000	1.31	Ax	4	2	1	An	34.40	138.00	2,090
Heinkel-Hirth	021	PJ	(2,000 hp) +800	9,000		Ax	4	2	1	An			2,860
Junkers Jumo	004	J	1,984	8,700	1.40	Ax	8	1	6	DF	34.00	152.00	1,585
Junkers Jumo	012	J	6,170	6,000	1.20	Ax	11	2	6	DF		204.00	4,400
Junkers Jumo	022	PJ	(4,500 hp) +1,600	6,000		Ax	11	2	6	DF			6,400

J = Turbojet
PJ = Propjet
AJ = Turbojet with thrust augmentation

RJ = Rocket-assisted turbojet
hp = Shaft horsepower
Cent = Centrifugal compressor

Ax = Axial compressor
DE = Double entry
SE = Single entry

RF = Reverse flow
DF = Straight-through flow
An = Annular combustion chamber

been included in this book to cover the various methods and procedures of testing.

Gas turbine development in the aviation field has come farther in a shorter time than the development in any other field. The Germans were first, the British were a close second, and we can say Americans came third. Table 1, originally published in the *Automotive and Aviation Industries* magazine and subsequently enlarged, shows the various units built here and abroad. Many of these were constructed by mass-production methods, and some gave an excellent account of themselves in World War II. At the end of World War I, when aviation first started in earnest, the reciprocating gasoline engine was the only source of aircraft power. Today, the jet or propeller-drive gas turbine unit is pushing rapidly forward. It is definitely replacing the conventional engine of the larger sizes and is going to powers not possible with the reciprocating engine.

Maintenance of the aviation gas turbine units cannot be compared with that of the heavy-duty equipment. Even if the gas turbine unit will last only 500 hours we may find that the plane in which it is installed lasts a much shorter time, especially if it is a fighter plane. For war purposes, speed and power are the primary requirements; life of equipment is purely secondary. From the lessons learned during World War II, we can make the jet and gas turbine propeller-drive planes conservatively enough for commercial purposes. The gas turbine life may still be shorter in hours than that of the reciprocating engine, but on a mileage basis, with the gas turbine jet or propeller drive at higher speeds, we will in time, no doubt, find the gas turbine unit requiring less maintenance per mile than the reciprocating engine of the same output.

PART I

HEAVY EQUIPMENT

Dr. Adolphe Meyer of Brown Boveri & Company Discussing That Company's Latest Revolutionary Gas Turbine Development Before the American Society of Mechanical Engineers in December, 1946.

Chapter II

Operating Experience with Gas Turbines[1]

The purpose of this chapter is to report on the operation and maintenance record of seven gas turbines installed by Sun Oil Company for the propulsion of axial air compressors in its oil refineries. The compressors supply high-pressure air used in connection with the catalytic cracking of petroleum for the manufacture of 100-octane aviation fuels and high-octane motor fuels.

It is assumed that those to whom this book is available are familiar with the theory of gas turbines and their general characteristics[2]; therefore these problems are not discussed. The commercial status of an important application of gas turbines, as indicated by operations since 1936, is appraised. The seven units installed in Sun Oil Company refineries during this time are described from standpoint of maintenance and reliability. The conclusion reached is that the installations have proved eminently satisfactory and have justified by performance their selection in preference to other types of units for the specific duties outlined.

General Facts

Prior to 1936, Sun Oil Company engineers, working in conjunction with the Houdry Process Corporation, adapted the Brown Boveri gas turbine and axial compressor to supply air in large quantities and at comparatively high pressures for

[1] Revision of report presented before the A.S.M.E., by Arthur E. Pew, Jr., vice-president, Sun Oil Company.

[2] For general characteristics of the gas turbine, refer to *The Modern Gas Turbine* by R. Tom Sawyer (2nd Ed. New York: Prentice-Hall, Inc., 1947)

catalyst regeneration in the Houdry process of catalytic cracking. At the Marcus Hook, Pa., refinery of Sun Oil Company, there was placed in operation, in the latter part of 1936, *the first successful large gas turbine unit in the United States* (Figs. 1 and 2.)

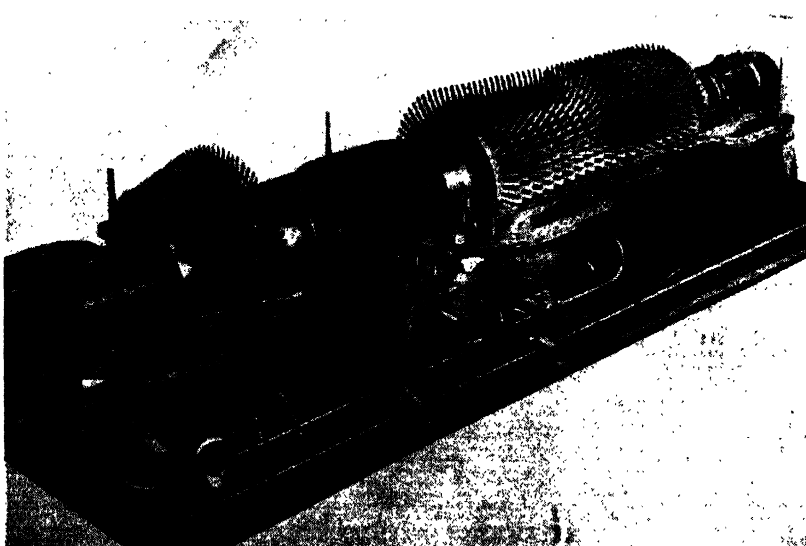
In the operation of the Houdry catalytic process for aviation and high-octane gasoline, it is necessary, for obtaining maximum efficiency, to maintain the catalyst in a state of high catalytic activity. For plants charging to the catalyst approximately 10,000 bbl per day, the air requirements are approximately 40,000 cfm at 45 psi (gauge). To obtain this quantity of air at this pressure, approximately 5600 hp are required.

A brief description of the Houdry catalytic cracking operation in the use of the gas turbine is necessary for an understanding of the performance records given hereafter in this book.

Oil is vaporized and passed through a catalyst. As one of the results of the chemical changes which occur, carbon is deposited on the catalyst. Since the efficiency of the catalyst is decreased by such deposits, the latter must be removed periodically. The catalytic containers are so arranged that carbon removal is on a continuous-cycle basis, certain containers being regenerated while others are operating on the oil cycle. The cycles from container to container are automatically controlled on a predetermined basis. The carbonaceous deposit is oxidized by the air from the compressor, and the products of combustion are passed to a gas turbine which in turn drives the axial compressor supplying air at 45 psi to the catalyst regeneration units. The heat content of the products of regeneration exceeds the energy necessary for the compression of the regeneration air. The excess is recovered as electric power and as steam for use in the process. In the usual installation, gas temperatures at the turbine inlet are 875 to 950° F and are exhausted at about 550° F. Inlet pressure to turbine is about 40 psi (gauge) and outlet pressure of compressor is 45 to 50 psi (gauge).

Courtesy of Sun Oil Company

Fig. 1. Sun Oil Company 40,000-cfm, 6600-hp Turbocompressor Installation with Motor Starter Installed in 1936 (Plant 11-4). This was the first large gas-turbine installation in the United States.



Courtesy of Sun Oil Company

Fig. 2. View of Brown Boveri Gas Turbine Compressor Unit of Fig. 1, with Top Cylinder Halves Removed. This compressor, installed in 1936 at Sun Oil Company's Plant 11-4 at Marcus Hook, Pa., is still in use.

Extent of Use

As of this date, there are in use in the United States 26 gas turbine units on Houdry cracking plants. Twelve units are of 23,000-cfm normal rating, 13 units of 40,000 cfm, and one unit of 60,000-cfm rating. Table 1 lists present installations in the United States.

TABLE 1

SUMMARY OF TURBOCOMPRESSORS
IN SERVICE ON STATIC-BED HOUDRY UNITS

Company	Location	No.	Nominal Rating Cfm	Mfg.
Sun (11–4)	Marcus Hook, Pa.	1	40,000	Brown Boveri
Magnolia	Beaumont, Tex., No. 1	1	23,000	Brown Boveri
Sun (12–3)	Marcus Hook, Pa.	1	40,000	Brown Boveri
Socony	Trenton, Mich.	1	23,000	Allis-Chalmers
Socony	Brooklyn, N. Y.	1	23,000	Allis-Chalmers
Sun	Toledo, Ohio	1	40,000	Allis-Chalmers
Magnolia	Beaumont, Tex., No. 2	1	40,000	Allis-Chalmers
Socony	Buffalo, N. Y.	1	23,000	Brown Boveri
Socony	E. St. Louis, Ill.	1	23,000	Brown Boveri
Socony	Augusta, Kan.	1	23,000	Allis-Chalmers
Socony	Paulsboro, N. J.	1	23,000	Brown Boveri
Sun (12–3 R	Marcus Hook, Pa.	1	[a]23,000	Allis-Chalmers
Sun (10–3)	Marcus Hook, Pa,	2	40,000	Allis-Chalmers
Tide Water	Bayonne, N. J.	1	60,000	Allis-Chalmers
Socony	E. Chicago, Ind.	1	23,000	Allis-Chalmers
Std. Oil Calif.	El Segundo, Calif.	1	40,000	Allis-Chalmers
Gulf Oil Corp.	Port Arthur, Tex.	1	40,000	Allis-Chalmers
Sun (15–1)	Marcus Hook, Pa.	1	40,000	Allis-Chalmers
Sinclair Ref.	Houston, Tex.	1	40,000	Allis-Chalmers
Magnolia	Beaumont, Tex., Nos. 3 and 4	2	23,000	Allis-Chalmers
Sinclair Ref.	Corpus Christi, Tex.	1	40,000	Allis-Chalmers
Southport Pet.	Texas City, Tex.	1	40,000	Allis-Chalmers
Std. Oil Ohio	Cleveland, Ohio	1	40,000	Allis-Chalmers
Std. Oil Ohio	Cleveland, Ohio	1	23,000	Allis-Chalmers

SUMMARY

Nominal Rating Cfm	*No.*
23,000	12
40,000	13
60,000	1
Total	26

[a] 23,000-cfm casing with blades cut for 16,000-cfm output.

There have been installed in Sun Oil refineries seven gas turbocompressor units, the first one in 1936 and the most recent in 1943. Table 2 lists pertinent data on these units.

TABLE 2

TURBOCOMPRESSOR INSTALLATIONS, SUN OIL COMPANY

Location	Cap. Std. Cfm	Rpm	Means of Starting	Date in Service	Type Thrust Bearing
M.H. 11–4[c]	40,000	5,180	Motor	Dec. 1936[a]	Angular-contact bal
M.H. 12–3[c]	40,000	5,180	Motor	July 1939[a]	
Toledo 3[b]	40,000	5,180	Motor	Aug. 1939[a]	
M.H. 12–3[b]	16,000	6,050	Turbine	Nov. 1940	
M.H. 10–3[b] N	40,000	5,180	Turbine	Dec. 1940[a]	
M.H. 10–3[b] S	40,000	5,180	Turbine	Jan. 1941[a]	
M.H. 15–1[b] S	40,000	5,180	Turbine	Aug. 1943	Kingsbury

All compressors 45-psi discharge pressure.
All units have 1,500-hp synchronous motor generators except that the 16,000 cfm has 500 hp.

[a] Angular-contact bearings removed. Kingsbury installed since 1943.
[b] Manufacturer, Allis-Chalmers
[c] Manufacturer, Brown Boveri.

Description of Sun Units

The gas turbocompressor units as used by the companies listed in Table 1 are all of the same general design other than capacity. The principal specifications are as follows:

Capacity compressor unit (based on 60° F), cfm	23,000 to 60,000
Suction pressure (based on 60° F)	atmospheric
Discharge pressure, lb (gauge)	45
Turbine gas quantities, 40,000-cfm size, lb per min	3170
Pressure, turbine inlet, lb (gauge)	40
Temperature, turbine inlet	800 to 950° F
Speed of set, rpm	5180
Weight of set, 40,000-cfm size, lb	70,000

The following equipment is required for the 40,000-cfm turbocompressor units as installed at the Sun Oil Company:

TABLE 3

EQUIPMENT
40,000-CFM TURBOCOMPRESSOR UNITS
SUN OIL COMPANY

Gas turbine	Brown Boveri or Allis-Chalmers	6200 hp, 950° F reaction-type five stages
Compressor	Brown Boveri or Allis-Chalmers	Axial-type 20 stages
Governor and oil pump	Brown Boveri or Allis-Chalmers	
Auxiliary oil pump	Brown Boveri, Roper, Allis-Chalmers	
3% overspeed valve	Brown Boveri, Allis-Chalmers, Republic	
10% overspeed valve	Brown Boveri or Allis-Chalmers	
Oil coolers	Andale or Griscom-Russell	
Oil filter	Nugent cloth filter (emergency) Shriver blotter press (normal opr)	
1500-hp syn motor	Allis-Chalmers	2300 v, 1792 rpm
Starting units	Motors, Allis-Chalmers or General Electric. Steam turbines, Moore	
Reduction gear unit	Moore	Between starter and 1500-hp motor
Reduction gear unit	Falk Herringbone or Brown Boveri	Between 1500-hp motor and turbocompressor 5166 to 1800 rpm

Materials

Table 4 shows the materials used in the construction of the principal parts of the turbo and compressor units.

TABLE 4

MATERIALS, TURBOCOMPRESSOR UNITS
SUN OIL COMPANY

Sealing strips	18 chrome, 8 nickel (originally were nickel)
Gas turbine blades	Stainless steel
Compressor blades	Nickel steel
Gas turbine rotor	Chrome-nickel steel forging
Gas turbine cylinder	Molybdenum cast steel
Compressor rotor	Steel forging
Compressor cylinder	Cast iron
Bed plate	Cast iron

Method of Operation

Figure 3 gives a splendid view of a typical Houdry process unit. Figure 4 shows diagrammatically the arrangement of a gas turbine blower generator and the starting motor or turbine interconnected with a catalytic cracking unit. In starting, the turbine is rotated at approximately 2000 rpm for several hours by means of the starting motor or starting steam turbine, while bearings are warmed up and equipment checked. During this time, the air delivery from the compressor is circulated through a by-pass to the turbine. After partial speed—that is, 25 to 30 per cent of normal, has been reached and the units checked, the temperature is brought up by the pressure burner preceding the turbine, Figure 4. The volume of air being circulated and the pressure is raised over a period of 8 to 10 min by gradually bringing the unit to its full speed of 5180 rpm. The heat of the discharge air is then gradually raised by the air heater located on the discharge side of the compressor, and the air thus heated is passed to the catalytic cases. Following this, the gases resulting from the regeneration of the carbonaceous deposits, and coming from the catalytic cracking case, supply the energy for the operation of the turbocompressor unit, and the pressure burners are shut off.

Under normal operating conditions, the amount of excess power supplied to, or developed by, the unit depends upon the type of catalytic cracking reaction being practiced—that is, with increase in carbon deposition, power will increase. In general, with air temperature to the compressor at 60° F. and with gas temperature to the turbine at 950° F. the 40,000-cfm units will generate up to 900 kw excess over required power for driving the compressor.

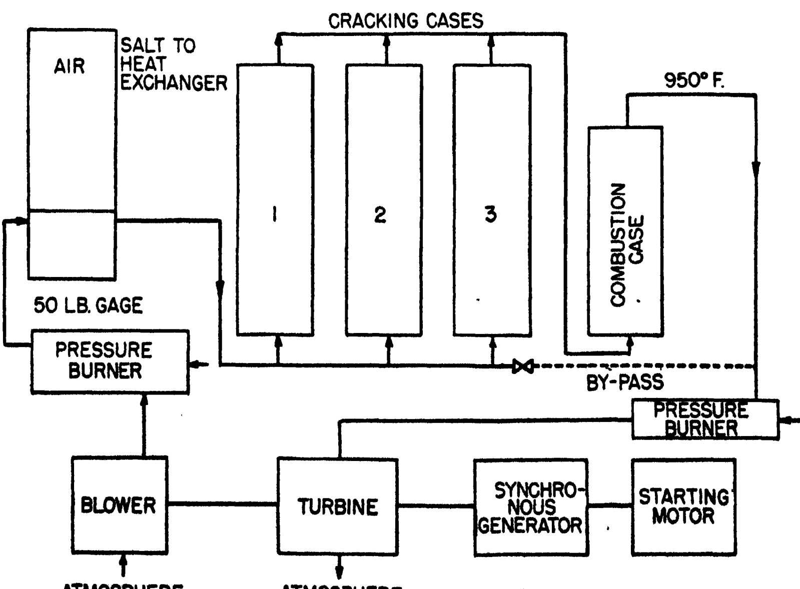
Service Record

Table 5 shows the number of operating days on each of seven Sun Oil installations in relation to the number of days shutdown on account of turbocompressor difficulties. The plants are listed in the order of their installation.

Under the heading *Days Off Account Turbo Trouble* is in-

Courtesy of Sun Oil Company

Fig. 3. View Showing a Typical Houdry Process Unit at the Sun Oil Company Refinery at Marcus Hook, Pa.



Courtesy of Sun Oil Company

Fig. 4. Air-flow Diagram.

cluded time chargeable to the complete turbocompressor unit and the equipment in connection therewith necessary for operation. The time shown represents that required for making repairs. It is believed by the operating division of Sun Oil Company that time off caused by turbine troubles and troubles in connection with turbines has not been excessive under the conditions encountered.

TABLE 5

SUN OIL COMPANY
SUMMARY TURBOCOMPRESSOR OPERATION
TO OCT. 31, 1945

Location Plant	Operating Days	Days Off Account Turbo Trouble	Total Days	% Time Turbo Trouble
Marcus Hook				
11-4	2750	50	2800	1.82
12-3 C	2033	29	2062	1.41
12-3 R	1537	45	1582	2.93
10-3 N	1542	37	1579	2.40
10-3 S	1510	29	1539	1.92
15-1 S	693	0	693	0.00
Toledo	2194	11	2205	0.50

Failures Causing Turbocompressor Outages

Table 6 shows failures causing outages in the order of their prevalence.

Table 7 classifies failures causing turbocompressor outages in relation to number of failures and days lost for each type of failure.

It will be noted that out of 53 failures, 32 were caused by bearing trouble. Out of 201 days lost time from all causes, 111 were due to failures of bearings. In this connection, Table 2 shows the type of bearings installed in Sun Oil units. An early discovery was that the angular-contact ball-bearing type of thrust bearing was not satisfactory for the conditions encountered, and as shown by the table, the angular-contact bearings have all been replaced with the Kingsbury type thrust bearing with the exception of one unit at Marcus Hook which will

TABLE 6

FAILURES CAUSING TURBINE OUTAGES
SUN OIL COMPANY

1. *Bearings*
   - Thrust bearings
   - Spring rings (roller bearings)
   - 1500-hp motor bearings
2. *Lubrication*
   - Dirt in oil sprays—causing bearing failures
   - Water in oil (condensate)—causing bearing failures
   - Main oil pumps—broken shaft spline as result of bearing failures
   - Auxiliary oil pump—failure to start when main pump failed
3. *Blades*
   - Loose blades—latest design with integral root, no loose blades in 22 months on one unit
   - Erosion—gas turbine blades
4. *Air seal strips* (failure does not necessarily cause shutdown)
   - Material
   - Rubbing—due to change in alignment
   - Expansion joints—harness
5. *Explosions*
   - 11–4 by-passing air into oil vapors—experimental case
   - 12–3 reformer—air by-passed into oil vapors
   - 12–3 cases—gas explosion—gas burner opened by mistake
   - 10–3 dirty oil burner—no atomization
6. *Miscellaneous*
   - Foreign objects in compressor or turbine
   - Failure to synchronize before cutting motor in when starting unit
   - Errors in maintenance or operation

TABLE 7

FAILURES CAUSING TURBOCOMPRESSOR OUTAGES
SUN OIL COMPANY

Location Plant	Bearings		Lubrication		Blades		Air Seals		Explosions		Misc.		Total	
	No.	Days	No.	Days	No.	Days	No.	Days	No.	Days	No.	Days	No.	Days
Marcus Hook														
11–4	18	43	—	—	—	—	—	—	1	5	1	2	20	50
12–3 C	4	12	—	—	1	4	1	3	1	2	2	8	9	29
12–3 R	4	31	3	14	—	—	—	—	—	—	—	—	7	45
10–3 N	3	15	3	14	—	—	—	—	1	5	1	3	8	37
10–3 S	—	—	—	—	3	15	—	—	—	—	2	14	5	29
15–1	—	—	—	—	—	—	—	—	—	—	—	—	0	0
Toledo	3	10	—	—	—	—	—	—	—	—	1	1	4	11
Total	32	111	6	28	4	19	1	3	3	12	7	28	53	201

be changed. There have been no thrust bearing failures or bearing repairs since the first Kingsbury type was installed on a Sun Oil unit in June 1942, and these bearings have been in satisfactory operation since that time. The other units were converted at various times in the interval. In view of the record of no failures, it is reasonable to assume that the Kingsbury type of bearing is satisfactory for these units. Had Kingsbury thrust bearings been used from the beginning, the total time lost probably would have been reduced approximately 55 per cent.

The tables show that lubrication difficulties caused a loss of 28 days and that six failures were due to faulty lubrication. Lubrication difficulties were caused by dirt in the oil sprays, condensate in the oil, broken shafts in main oil pumps arising from bearing failures, and failure of auxiliary oil pumps to start when the main pump was out of operation. Lubrication difficulties probably have been largely overcome by the installation of blotter presses because there have been no lubrication failures since these presses have been installed.

Because of failure of blades, either the compressor or the turbine was responsible for four shutdowns of 19 days lost time. Loose blades are caused by a defect in an original design in which the blades were not integral with the roots. Later designs by Allis-Chalmers have provided an integral root blade, and no trouble has been observed in over 27 months operation in a Sun Oil unit with this type of blade. Some trouble has been given by erosion of the gas turbine owing to the presence of fine particles of catalyst or small deposits of carbon dust in the gases. Experimental work is being done on how to eliminate this condition by trapping the air entrances to the turbine. However, the erosion of the gas turbine has not been serious enough to become an important problem.

The tables indicate one shutdown of three days duration for failure of air seals. Air seal strips are subjected to rubbing due to change in alignment, and gradually wear out. However, the seals may be repaired during a normal shutdown without excessive difficulty and are not considered as being a weakness which might necessarily cause shutdowns. However, improve-

ments in air seals become more important as shutdowns for other causes are eliminated.

There have been three shutdowns of 12 days total duration due to explosions. These explosions are not due to turbine design, but are the result of errors in operation or failures in handling other parts of the equipment services by the turbines. None of the three explosions was of sufficient intensity to rupture the equipment.

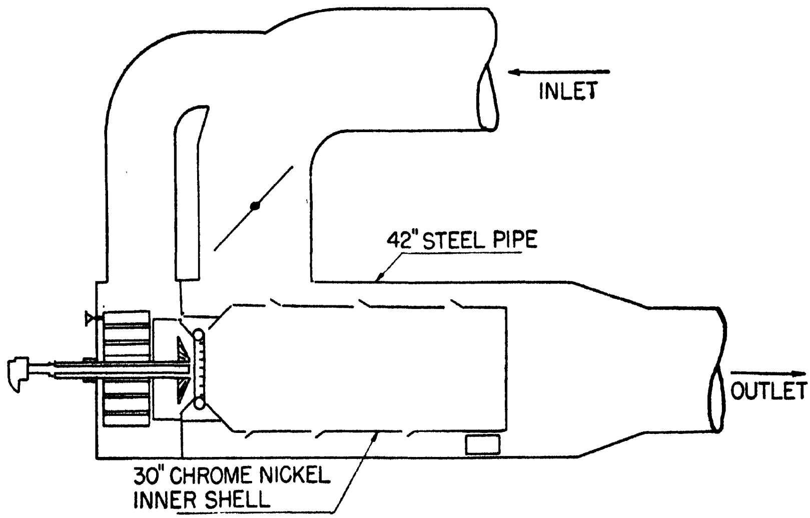
Under the column headed *Miscellaneous,* there were seven shutdowns of 28 days duration due to minor turbine troubles and errors in maintenance or operation. In one case, a foreign object, left in the air compressor, caused the blading to be stripped.

Summarizing, bearing, lubrication, and blading failures are the three principal causes of shutdowns. They accounted for 158 days lost time out of the total of 201, and for 42 shutdowns of the total of 53. The changes described above probably have satisfactorily and substantially eliminated these causes of shutdowns.

Problems of Installation

Probably the most important consideration with regard to installation arrangement is ample provision in the piping to reduce piping strains to a minimum. The piping to the turbine and compressor units is unusually large; the air inlet to the compressor on the 40,000-cfm machine is 36 in. and the air outlet, 30 in. The gas inlet to the turbine requires a 30-in. pipe, and the exhaust, a 36-in. pipe.

Because of the high temperatures of the piping to the turbine, provision must be made to handle the expansion and contraction, since the machine is relatively light in construction and the speeds high, which require that the alignment be accurately maintained. The design of the burners used for starting the turbocompressor requires careful consideration. Figure 5 shows the arrangement of the combustion chambers, one ahead of the air heater and the other ahead of the turbine. Figure 6 is a photograph of the two combustion chambers that are represented diagrammatically in Figure 5.



Courtesy of Sun Oil Company

Fig. 5. Arrangement of Combustion Chamber.

Courtesy of Sun Oil Company

Fig. 6. Combustion Chambers of Fig. 5 Used in a Sun Oil Houdry Process Unit. Shown are burners, air-discharge line from compressors, lines to catalytic cases, and return lines to gas turbines.

Air at a temperature of approximately 350° F and 45 lb pressure from the outlet of the air compressor enters the combustion chamber through a 30-in. pipe. The air flow is divided at this point, a portion passing through an air register and thence to the oil burner for combustion. The balance of the inlet air passes through a control damper and around the outside of a 30-in. chrome-nickel shell inserted in the 42-in. steel pipe. The arrangement of perforations on the inner shell allows a pattern of air infiltration that creates a swirling and mixing action within the inner shell which provides for cooling of the shell walls. The products of combustion and the excess air coming at the outlet of the 42-in. pipe section and pass therefrom to the catalytic cases for regeneration of the catalyst. The temperature of the mixture is approximately 950° F and the temperature within the inner shell is approximately 2000° F. The air capacity of the burner is approximately 40,000 cfm at 45 lb pressure. No refractory material is used. This type burner has operated for a number of years with entirely satisfactory results. The design is an adaptation by Sun Oil engineers from the basic design used by the Brown Boveri Corporation.

Arrangements of valves, control instruments, and so forth, have been worked out on the basis of experience and have given satisfactory results. Detailed description of the installation is too lengthy to be incorporated in this chapter. However, it may be said that the installations are not complex and involve no special operating or maintenance difficulties.

Maintenance Costs

The cost of maintenance and repairs of the various units listed in this report have varied from a high of approximately $1000 a month to a low of $300. Maintenance and repairs may be conservatively estimated as not in excess of 3 per cent per year of capital investment. The most costly repair is the reblading of the gas turbine, estimated at $13,000, of which approximately $12,000 is for blades. The cost of cleaning the units has averaged about $300 per year each.

Summary

The operating record of the gas turbocompressor units of Sun Oil's plants have been such as to justify the continuation of the use of this type of equipment in future installations. The cyclic operation of the Houdry units requires the steady flow of regeneration air under varying pressure differentials and requires also changes in the system pressure differentials caused by changes in operating conditions. The quantity of air delivered by the axial-flow compressors fluctuates less than one per cent for 20 per cent variation in discharge pressure as contrasted with approximately 20 per cent or more for similar changes in pressure conditions in the case of centrifugal compressors. No suitable centrifugal compressor or prime mover that meets the requirements of economy and efficiency necessary in connection with the regeneration of catalysts in the Houdry catalytic operation has as yet been offered.

In the application under discussion, it is necessary at high pressures and in large quantities to handle gas and to utilize combustion products resulting from the chemical reactions obtained from the catalytic cracking processes. A catalytic plant requiring 40,000 cfm at 45 lb pressure could not obtain high economic efficiency were it necessary to compress this amount of air without using resultant combustion products as the source of power for the compression of the air.

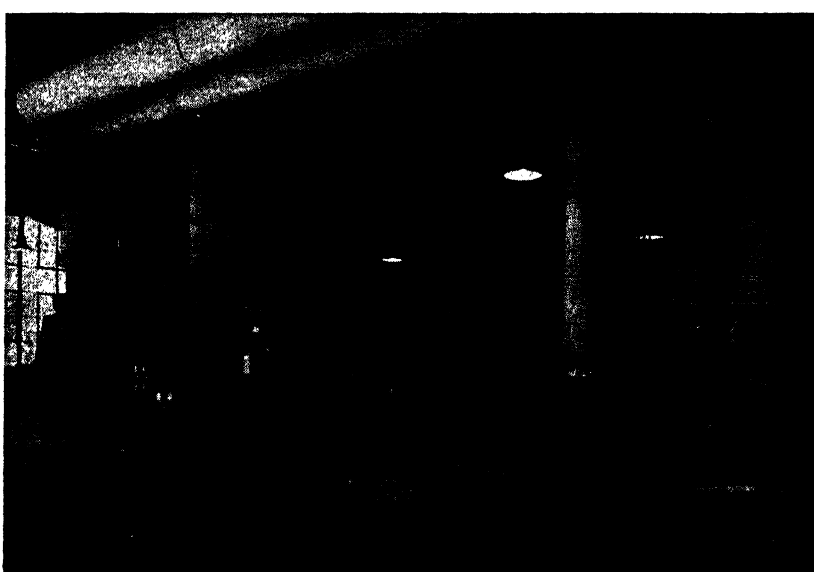
Chapter III

Houdry Gas Turbine Units

General Description

The unit consists of an axial compressor direct-connected to a gas turbine at one end and coupled to the high-speed shaft of a reduction gear at the other end. A starting turbine or motor is connected to the low-speed shaft of the gear, and on some units, a generator is connected to the same shaft between the gear and the starting unit (Fig. 1).

The compressor is used to supply compressed air to a process that discharges hot gas as a by-product, which is used



Courtesy of Sun Oil Company

Fig. 1. Sun Oil Installation of a 40,000-cfm, 6600-hp Turbocompressor with Steam Turbine for Starting.

to operate the driving turbine. When a generator is used, it absorbs any power developed by the gas turbine in excess of the amount required to drive the compressor. When the generator is not furnished, excess power must be dissipated by blowing off air or hot gas at some point between compressor discharge and turbine inlet.

The speed of the unit is controlled by an overspeed protective governor driven from the low-speed shaft of the gear. The governor opens a blowoff valve on the blower discharge or turbine inlet whenever the speed gets too high.

An emergency overspeed governor is located at one end of the gas turbine. In the event of excessive speed, this governor trips a by-pass valve on the turbine.

The lubrication-and-governor oil system consists of one common oil tank, two gear-type oil pumps driven from the low-speed shaft of the reduction gear, and a motor-driven auxiliary oil pump. The auxiliary oil pump is used for starting the unit, and is equipped with automatic controls to put it in operation when either of the main oil pumps should fail to maintain its proper pressure.

List of Equipment

Compressor

The compressor is an Allis-Chalmers Type VA-820, 20-stage axial compressor rated 45 psi (gauge) at 5180 rpm when handling 40,000 cfm of inlet air at 60° F and 14.7 psia. The compressor is equipped with roller bearings and Kingsbury thrust bearing.

Gas Turbine

The turbine is an Allis-Chalmers gas turbine rated 6200 hp and 5180 rpm, and designed to operate on gas at 38 lb (gauge), 950° F maximum, with atmospheric exhaust. The turbine is equipped with roller bearings, emergency overspeed governor and safety by-pass valve.

Reduction Gear

The gear is a Falk 8 HQP–2 Speed Reducer, with an input speed of 5166 rpm and an output speed of 1800 rpm (1500 rpm

for 50-cycle units). The pre-emergency governor and two gear oil pumps are enclosed in the gear housing and are driven from the low-speed shaft of the gear.

Lubrication

The gas turbine bearings, compressor bearings, and gear are lubricated by the main oil pump, rated 65 gpm at 20 lb, driven from the low-speed shaft of the reduction gear, and taking oil from a 157-gal oil tank. The system contains an Andale oil cooler and a Roper motor-driven rotary auxiliary oil pump with built-in relief valve, rated 50 gpm at 60 lb. Mercoid controls are provided to start the auxiliary oil pump whenever the main oil pump or the governor oil pump should fail to maintain its pressure.

Regulation

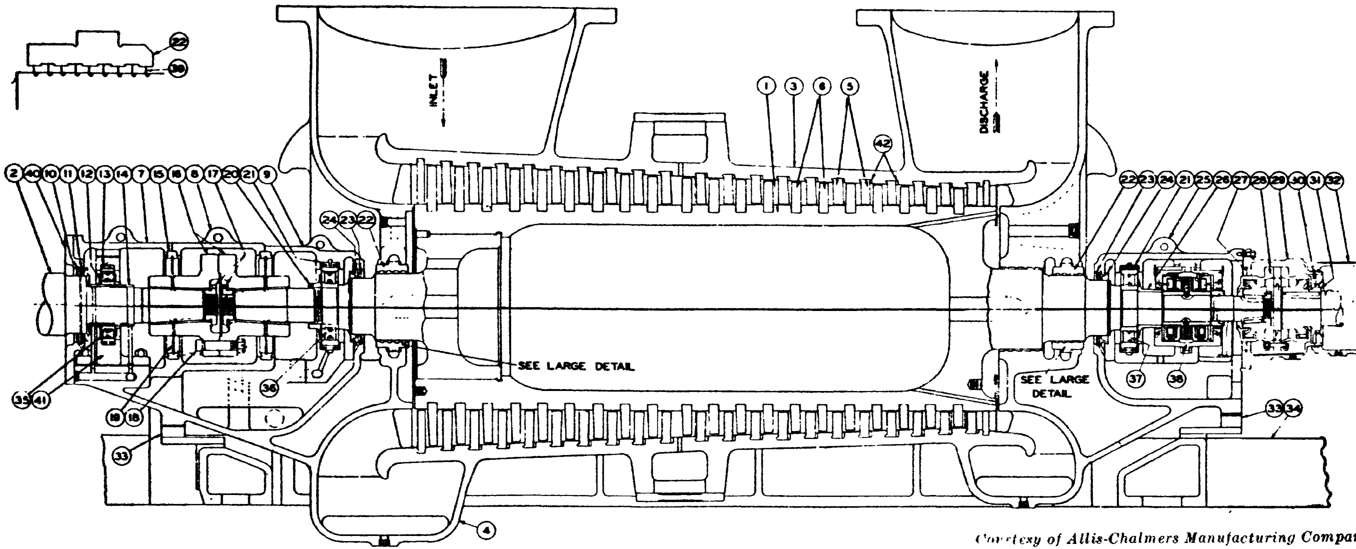
The speed of the unit is limited by an Allis-Chalmers Vertical Mechanical Speed Governor driven from the low-speed shaft of the speed reducer and arranged to function as a pre-emergency speed-limiting governor. By means of an oil relay, this governor operates a Smoot eight-inch oil-operated blow-off valve. Blowing off air (or hot gas) reduces the speed of the unit. The gas turbine is equipped with an Allis-Chalmers type emergency overspeed governor arranged to trip an oil relay which opens the safety by-pass valve on the turbine at approximately 10 per cent overspeed. Oil for the governor oil relays is furnished by an oil pump, rated 12 gpm at 60 lb, and driven from the low-speed shaft of the speed reducer.

A generator and starting turbine (or motor) are required.

Description of Compressor

Casing

The casing consists of a cast-iron cylinder, horizontally split and with inlet and outlet openings directed vertically upward and cast together with the upper half. The casing has 22 rows of stationary guide vanes which are caulked in place and held in grooves machined in the casing, following general steam turbine practice (Fig. 2).



Courtesy of Allis-Chalmers Manufacturing Company

1—Blower Spindle
2—Turbine Spindle
3—Cylinder (Upper Half)
4—Cylinder (Lower Half)
5—Cylinder Blading
6—Spindle Blading
7—No. 2 Bearing Cover
8—Coupling Cover
9—No. 3 Bearing Cover
10—Shaft Sealing Ring
11—Oil Baffle
12—Bearing Retaining Ring
13—No. 2 Bearing Support Ring
14—Bearing Shaft Sleeve
15—Sealing Ring Retainer
16—Coupling Assembly (Between Blower and Turbine)
17—Lock Nut
18—Fitted Bolt
19—Coupling Sealing Ring
20—Bearing Lock Nut
21—Bearing Support Ring
22—Sealing Strip Gland
23—Shaft Sealing Ring
24—Oil Baffle
25—No. 4 and No. 5 Bearing Cover
26—Bearing Lock Nut
27—Sealing Ring
28—Fast's Coupling
29—Coupling Housing
30—Oil Baffle
31—Coupling Spacing Ring
32—Coupling Housing Adapter
33—Cylinder Guides
34—Bed Plate
35—No. 2 Bearing
36—No. 3 Bearing
37—No. 4 Bearing
38—No. 5 Bearing (Thrust)
39—Sealing Strips
40—Cylinder End Cover
41—No. 2 Bearing Pedestal and Cap
42—Filler pieces and wedges

Fig. 2. Section through Axial Compressor.

Spindle

The rotor is of forged steel and consists of two parts—a drum with one shaft end, and a shaft end having a hollow cylinder for the rotor proper. The hollow cylinder is shrunk onto the drum and locked. The rotor is grooved to receive the rotor blading, which is wedged and caulked in place in a manner similar to that used for the cylinder blading. Labyrinth sealing strips are provided on the rotor ends where the shaft emerges from the casing.

Base Plate

The unit is mounted on a base plate of cast iron. The compressor and turbine are mounted in such a manner that the cold end of each machine is fixed and the other end is free to expand under the influence of heat.

Roller Bearings

The Nos. 1, 2, 3, and 4 bearings are roller bearings with bronze retainers that have more internal clearance between the rollers and the races than the standard type. The inner races have been given a stabilized heat treatment to prevent growth (Fig. 2).

The outer races of the Nos. 1 and 2 bearings fit in spring rings of a special alloy steel which are mounted in grooves machined in the turbine bearing covers. The spring rings aid in aligning the bearings and allow for radial expansion. The inner races are pressed on shaft sleeves and are locked by retaining rings.

The No. 3 bearing consists of either one or two duplicated roller bearings, as may be required to carry the load. The inner race has a shrink fit on the shaft and is locked against a shoulder by a lock nut. The outer race is mounted in a spring ring which fits in a groove in the blower low-pressure end bearing cover. On some compressors, where two duplicate bearings are used, the spring rings are mounted in a bearing retaining ring which separates the two bearings.

The No. 4 bearing is a duplicate of the No. 3 bearing except that in some cases the method of locking is different.

Kingsbury Style JJ-7 Thrust Bearing

Principal Parts

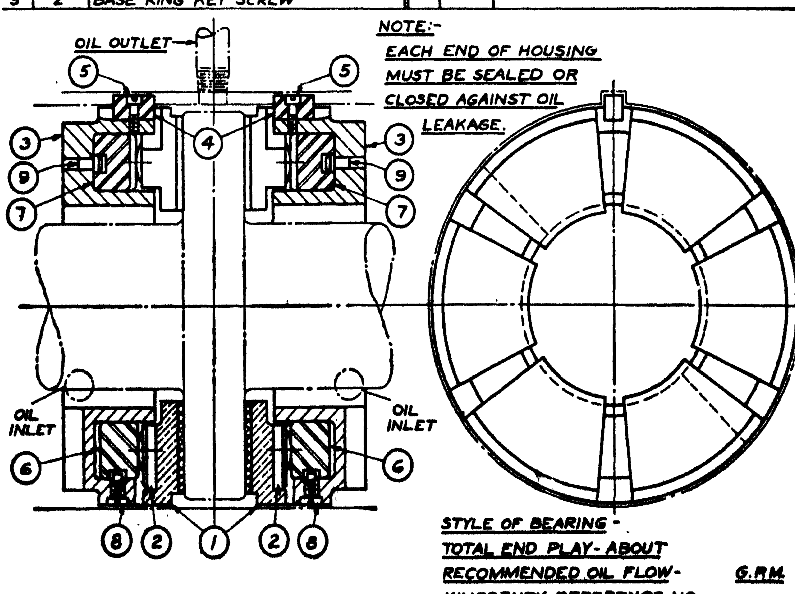
Figure 3 shows the general construction of bearing No. 5. The principal parts of the bearing include the rotating collar; the stationary pivoted segments, called *shoes;* and the load-equalizing mounting for the shoes. In the back of each shoe, a hardened steel pivot allows slight tilting in any direction.

In six-shoe bearings, the shoe loads are equalized and the bearings self-aligned by the leveling plates.

Cleaning and Inspection

As packed for shipment from the factory, the bearing surfaces are carefully protected against bruises, scratches, and

ITEM	REQ. FOR ONE BRG.	NAME	ITEM	REQ. FOR ONE BRG.	NAME
1	12	SHOE (BABBITT FACED)	6	12	UPPER LEVELING PLATE
2	12	SHOE SUPPORT (IN SHOE)	7	12	LOWER LEVELING PLATE
3	2	BASE RING (IN HALVES)	8	12	LEVELING PLATE SET SCREW
4	2	BASE RING KEY	9	12	LEVELING PLATE DOWEL
5	2	BASE RING KEY SCREW			



Courtesy of Allis-Chalmers Manufacturing Company

Fig. 3. Kingsbury Thrust Bearing.

corrosion. They are slushed with a neutral waterproof coating, and no wood or damp packing material is allowed to touch them. Subsequent damage may occur in reshipment or storage unless the same precautions are observed.

All parts of bearings, housings, and oil piping should be taken apart and thoroughly cleaned before assembling. Anti-rust coatings are removed with gasoline or kerosene. Rags or cloth are used for cleaning because waste leaves lint, which clings to minute burrs and may cause trouble in the bearing.

Important: A poorly cleaned bearing will score and wear out rapidly. A bearing surface is clean only when a white cloth that is wiped over it is neither stained nor soiled.

All bearing parts should be inspected after cleaning. All bruises on the babbitt faces are removed with a scraper. Slight bruises or rust on journal or collar surfaces are taken out with a fine oil stone. High spots caused by heavier bruises may need filling or scraping, but finishing should always be done with an oil stone. Deep rust requires refinishing.

The thrust collar must be exactly square with the shaft. If the collar is separate, bruises on the shaft shoulder should be removed before assembling.

Assembling

Any solid oil-seal rings used must be assembled over the end of the shaft in proper sequence.

Six-shoe bearing base rings are usually split for radial assembling. To assemble the split six-shoe type, the lower half, with its leveling plates, is first put over the shaft, and rotated into bottom position. The upper half is placed on the shaft in a similar manner, care being taken that the ends of leveling plates interlock properly. Retaining screws or spring clips are furnished to hold the leveling plates. Then all shoes are inserted, the base ring being rotated as needed. The halves should be bolted together if bolts are provided.

All bearing surfaces are oiled during assembly. The upper half of each base ring has a key that enters a keyway in the upper half of the housing. Force is never used during assem-

bling; parts that do not go together easily indicate that something is out of place.

A two-shoe bearing goes on the more lightly loaded side of collar. Because the base of such a bearing is in halves that are keyed at the joints, the shoes should be assembled radially as for six-shoe bearings.

The leveling washer, base ring, and (in standard types) the shoe cage of the three-shoe bearings must be assembled in proper sequence over the end of the shaft. The shoes are assembled radially.

If the shaft is threaded to hold the collar, a heavy spanner is used to drive the collar nut very tight.

End Play

To allow for oil films between the bearing surfaces and for expansion by heat, it is necessary to provide end play. End play increases with bearing size and shaft speed. Figure 3 specifies the desired amount.

End play is usually determined by filler rings that are solid or split according to bearing requirements. To adjust end play, the filler pieces are shimmed or machined as needed, care being taken to secure any split shims to prevent overlapping of edges of halves. Axial location on the shaft is sometimes controlled by filler rings.

End play is best checked by jacking the shaft fore and aft and measuring accurately the change in position. The housing cap must first be doweled or bolted in its exact position.

Lubrication and Cooling

These bearings are intended for forced lubrication and outside cooling. The bearing housing should be filled with oil before starting. The rate of circulation is specified approximately in Figure 3. Circulation should be such as to keep the outlet temperature within about 15° F above the inlet temperature (unless otherwise specified in the print). The pressure drop through the bearing is negligible. Oil pump and piping should be designed to handle the required flow.

Oil flow is restricted to specification by throttling the dis-

charge. Should this cause excessive leakage past end seals, the inlet should also be throttled.

The oil is cooled, usually by oil coolers supplied with water, but sometimes, for low-speed service, merely by radiation from the piping. Air cooling can be improved by a fan or blower.

Grade of Oil

Correct viscosity is important. Usually the thrust bearing has been selected with a view to using the same oil on it as on the rest of the machine. Special requirements will be noted on Figure 3 and on the Kingsbury name plate.

Changing to an oil much lighter than originally intended may cause the lubricating films to become dangerously thin. A much heavier oil will needlessly increase friction.

The oil must be clean and free from grit and other injurious substances. Fine grit has a scouring action. Poor oil may cause corrosion, sludge, or excessive evaporation.

Operation

The only attention normally required is to maintain proper circulation of clean, cool oil.

Since the bearing surfaces when running are completely separated by oil, there is practically no wear; hence no take-up is provided. The original scraper marks on thrust shoes in the larger sizes are visible, as a rule, even after years of service. These marks may reasonably be expected to remain visible if the bearing is clean when installed and if it is kept supplied with clean oil of proper viscosity.

Inspection and Replacement

For inspection (a) lift housing cap and (b) rotate base rings or shoe cages, and withdraw shoes radially. Shoes of larger sizes have tapped holes as aids for lifting.

Six-shoe base rings are usually split in order that they might be lifted out in halves. Other bearing parts should be removed over the end of shaft. If six shoes are used on one

side of collar only, that side should be noted for correct replacement.

Thrust collars have a sliding fit on shaft, and are readily removable after the nut is taken off. To draw off collars of the larger sizes, the tapped holes near the bore are used. Care should be taken not to bruise the collar.

When replacing the collar, the nut locks are relocated after retightening collar nut.

Repairs and Service

In case the shoes are rebabbitted or the collar refinished, the following precautions should be observed:

1. The collar must run perfectly square with the shaft. Tool or grinding marks are removed by lapping.

2. High-tin babbitt is used for the shoes. The latter are scraped to a surface plate after machining, and the radial edges are rounded slightly.

Gas Turbine—General Description

Figure 4 shows a sectional view of the gas turbine.

The turbine is of the straight-reaction type with five rows of stationary and five of moving blading.

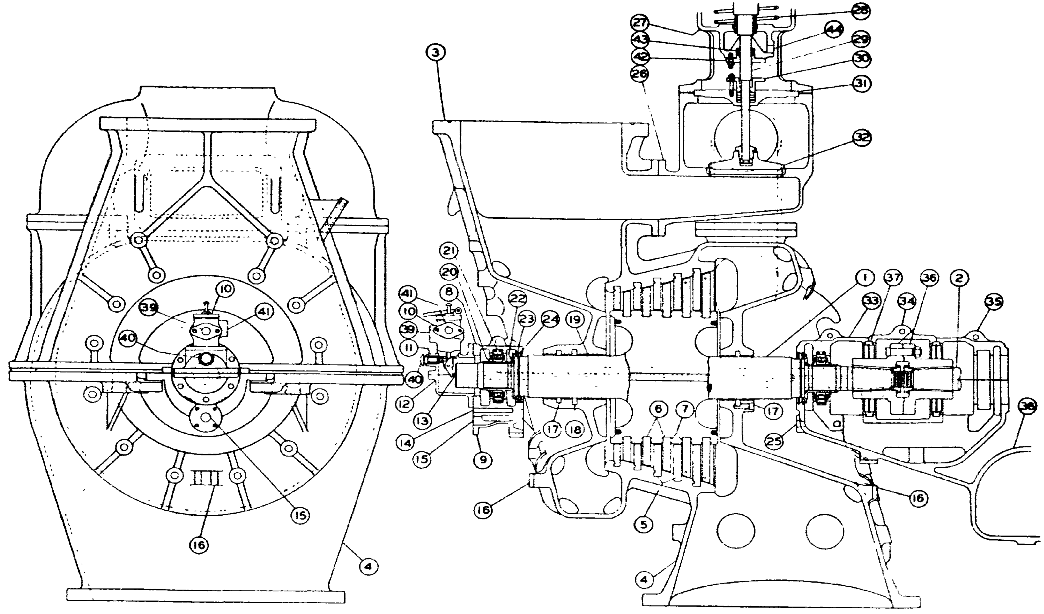
The turbine frame, or casing, is made of molybdenum cast steel and is split at the horizontal center line (Figs. 5 and 6). Both inlet and outlet nozzles are located in a vertical plane. The inlet nozzle is cast in one piece with the upper cylinder half, and the gas exhaust nozzle is cast in one piece with the lower cylinder half. A by-pass connection is provided to connect the inlet and the exhaust through a safety valve.

The casing is provided with a sufficient number of stay bolts and ribs, which are designed to maintain the shape of the casing yet permit it to expand freely.

The casing contains five rows of stationary blading made of stainless steel (Fig. 7). Figure 5 shows the cylinder casing finished-machined and grooved, and Figure 6, the cylinder casing with the blading inserted.

The cylinder casings used on turbines Nos. 10,033 and 10,034 are identical with that shown in Figures 5 and 6 with

1—Turbine Spindle
2—Blower Rotor
3—Cylinder (Upper Half, Containing Inlet Nozzle Connection and By-Pass Connections)
4—Cylinder (Lower Half, Containing Exhaust Nozzle)
5—Cylinder Blading
6—Spindle Blading
7—Spacer Pieces
8—#1 Bearing Housing (Upper Half)
9—#1 Bearing Housing (Lower Half)
10—Emergency Governor
11—Governor Housing (Upper Half)
12—Governor Housing (Lower Half)
13—Overspeed Governor
14—#1 Bearing Lubrication Inlet
15—#1 Bearing Lubrication Drain
16—Cylinder Guide
17—Sealing Air Inlet Duct
18—Leak Off
19—Gland Grooves
20—Bearing Retaining Ring
21—Bearing Support Ring
22—Bearing Shaft Sleeve
23—Oil Baffle
24—Shaft Sealing Ring
25—#2 Bearing Lubrication Inlet
26—Oil Control Valve Body
27—Oil Control Cylinder
28—Spring
29—Valve Spindle



Courtesy of Allis-Chalmers Manufacturing Company

30—Stuffing Box Gland (Lower)
31—By-pass Valve Cover
32—Valve Disc
33—#2 Bearing Cover
34—Coupling Cover
35—#3 Bearing Cover
36—Coupling Assembly
37—Sealing Ring Retainer
38—Blower Casing
39—Governor Drain Connection
40—Hand Trip
41—Governor Oil Inlet Connection
42—Stuffing Box Gland (Upper)
43—Packing
44—Auxiliary Drain

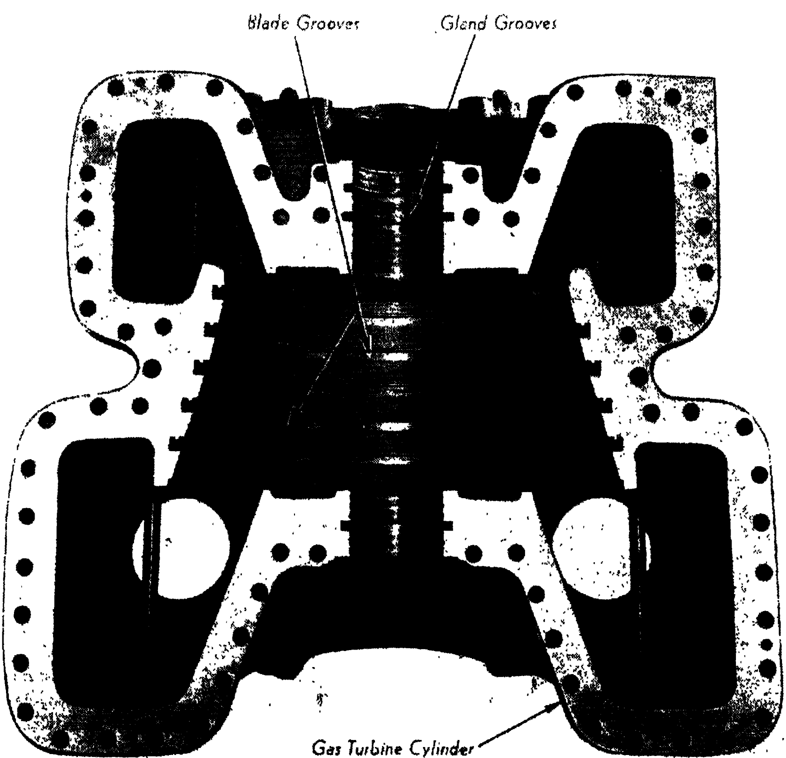
Fig. 4. Gas Turbine Cross Section.

the exception of a slight change in the cylinder-casing outline.

The spindle, or rotor, consists of a solid chrome-nickel steel forging (Fig. 8). Five serrated grooves are machined to receive the moving blading, which also is made of stainless steel (Fig. 7). Figure 9 shows the spindle in finished condition.

Sealing Glands, Bearings, and Coupling

Labyrinth glands are provided where the spindle ends pass through the casing (Fig. 4). Sealing air is taken from the blower exhaust and injected at a suitable point along the labyrinth in order to prevent gas from leaking to atmosphere. Separate valves are provided in the sealing lines to both glands.



Courtesy of Allis-Chalmers Manufacturing Company

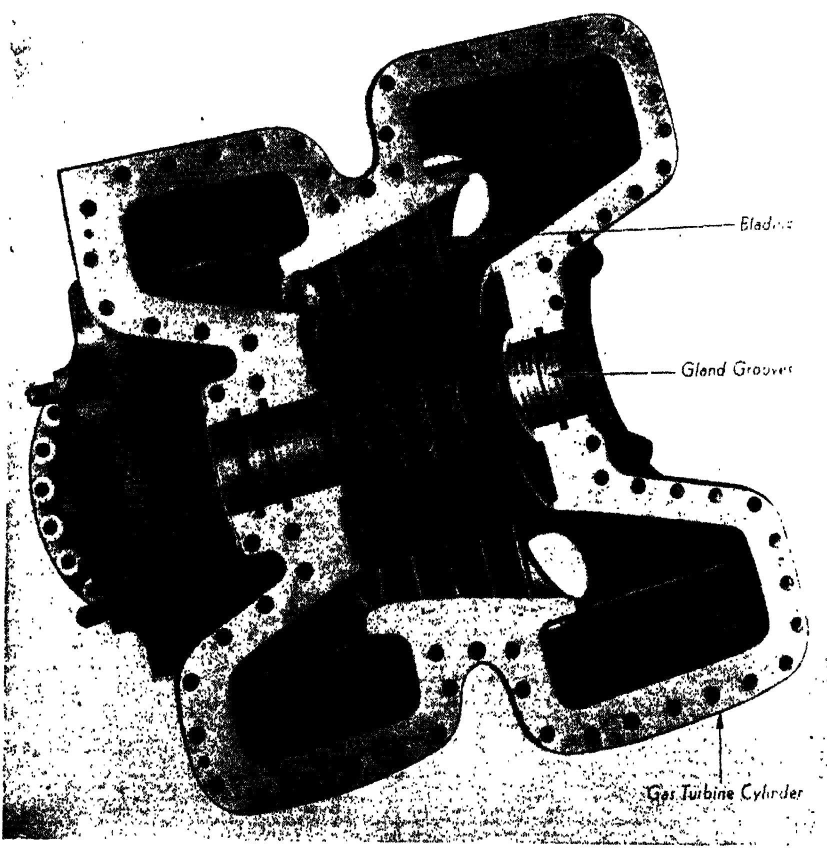
Fig. 5. Gas Turbine Frame without Blades.

The turbine rotor is carried by two special roller bearings. Lubricating oil is supplied to individual spray nozzles at a pressure of 20 to 30 psi.

A stiff coupling is provided to transmit the torque from the turbine to the blower.

By-pass Safety Valve

In order to protect the gas turbine from overspeeds above 10 per cent in case the electrical load is suddenly removed, a by-pass safety valve is provided. This valve, when actuated by the emergency stop, permits the gases to pass directly from

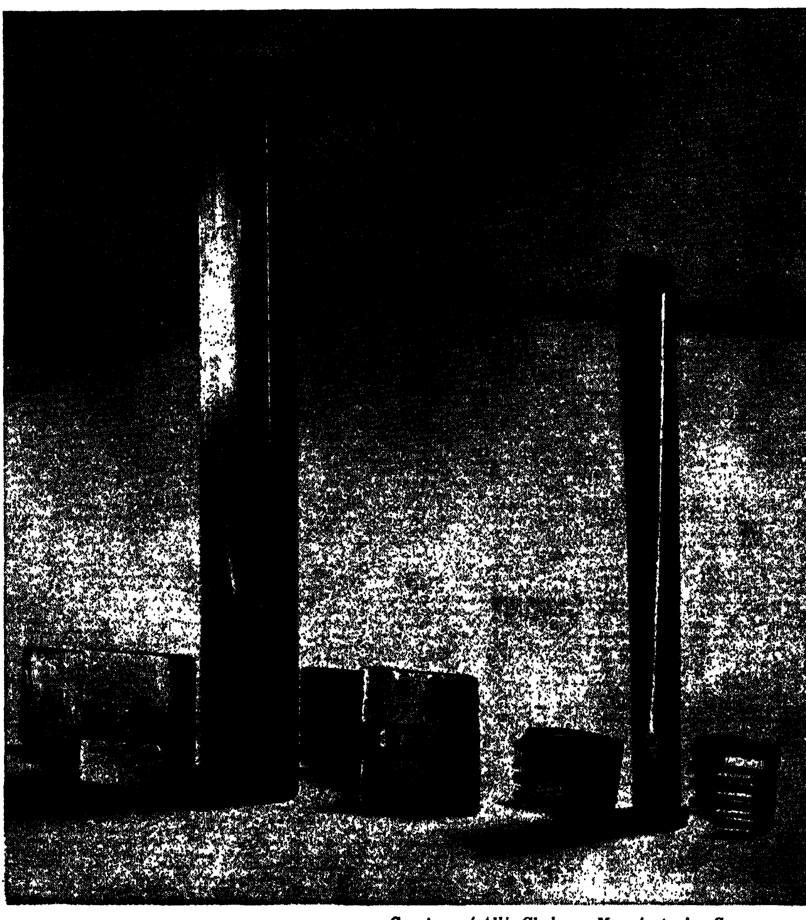


Courtesy of Allis-Chalmers Manufacturing Company

Fig. 6. Gas Turbine Frame with Blades.

the turbine inlet to the exhaust, thereby depriving the turbine blading of its motive fluid. Figure 10 shows the by-pass valve.

The by-pass valve, which is of the single-seated unbalanced type, is provided with a spring-loaded power piston (13). The unbalance of the valve is in the opening direction. The downward thrust of the piston (depending on the size and the oil



Courtesy of Allis-Chalmers Manufacturing Company

A B

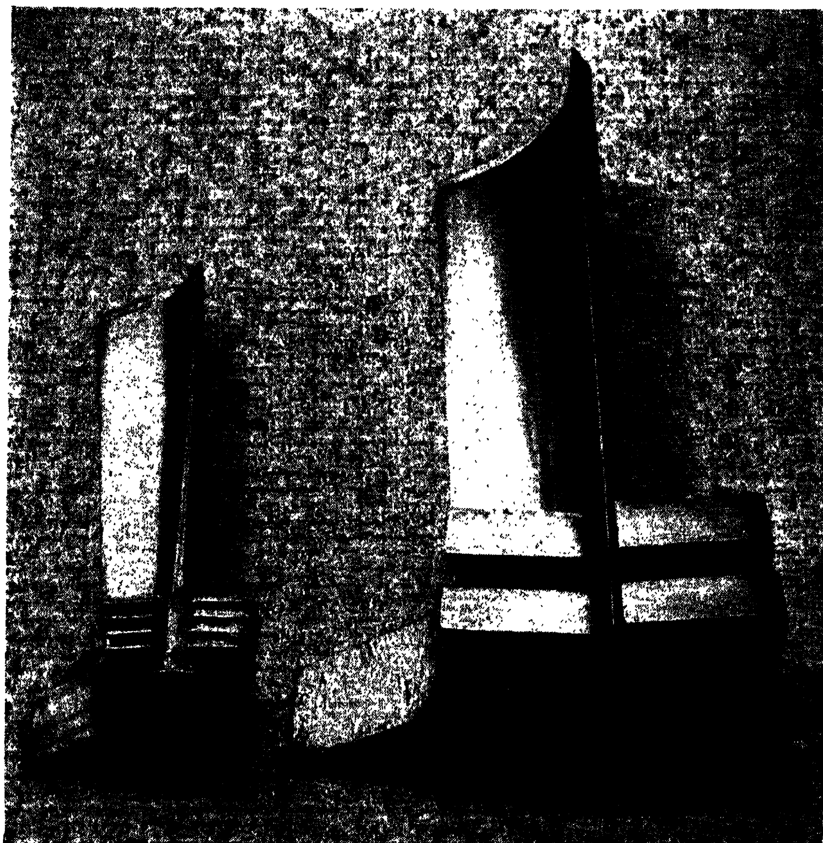
A—Long Stationary Blade with Spacer Pieces
B—Long Rotor Blade with Spacer Pieces

Fig. 7. Gas Turbine Blading.

pressure) is such that the valve is kept shut during normal operation.

Control oil at a pressure of 60 to 95 psi is supplied to the dome (18).

An orifice larger than opening *A* (21) supplies oil on top of the spring-loaded pilot valve (16). The pressure in the dome (18) will build up until the supporting spring (15) on



Courtesy of Allis-Chalmers Manufacturing Company

C D

C—Short Rotor Blade Assembled with Spacer Pieces
D—Short Stationary Blade Assembled with Spacer Pieces

Latest designs have the spacer pieces replaced by an integral root to the blade.

Fig. 7. Gas Turbine Blading (Cont.).

the pilot is compressed, and the pilot valve is seated. From that moment on, oil supplied to main piston (13) through opening *A* (21) cannot escape, and the piston travels down-

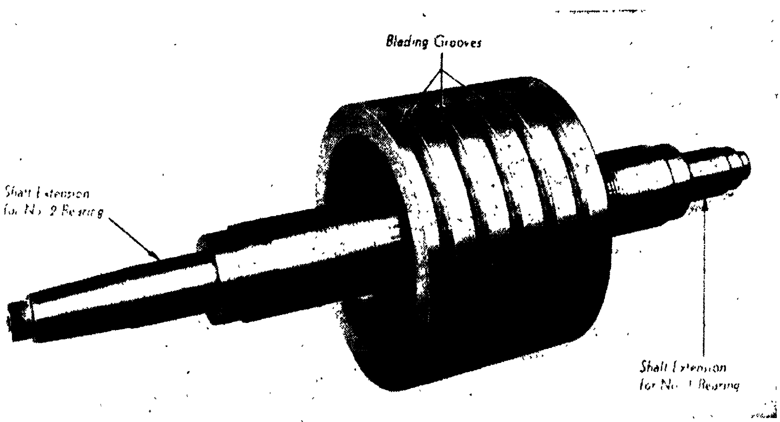


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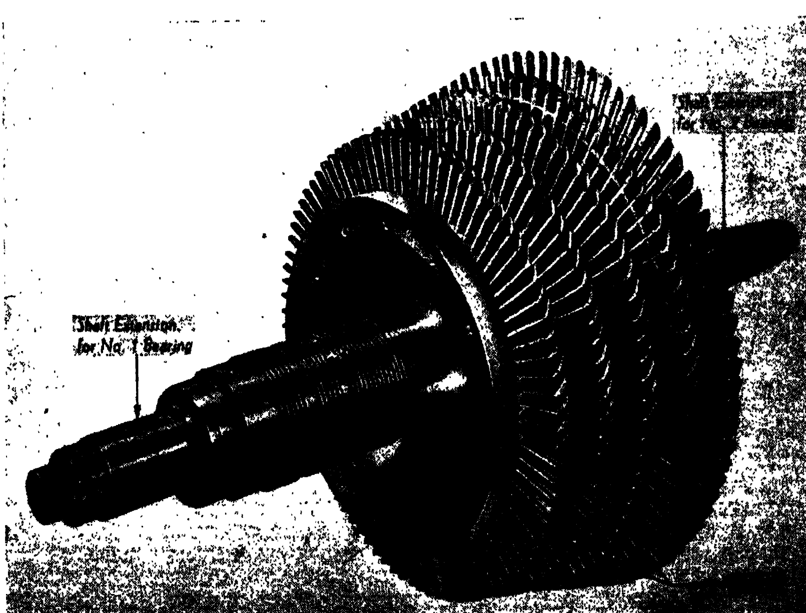
Fig. 8. Gas Turbine Rotor without Blades.



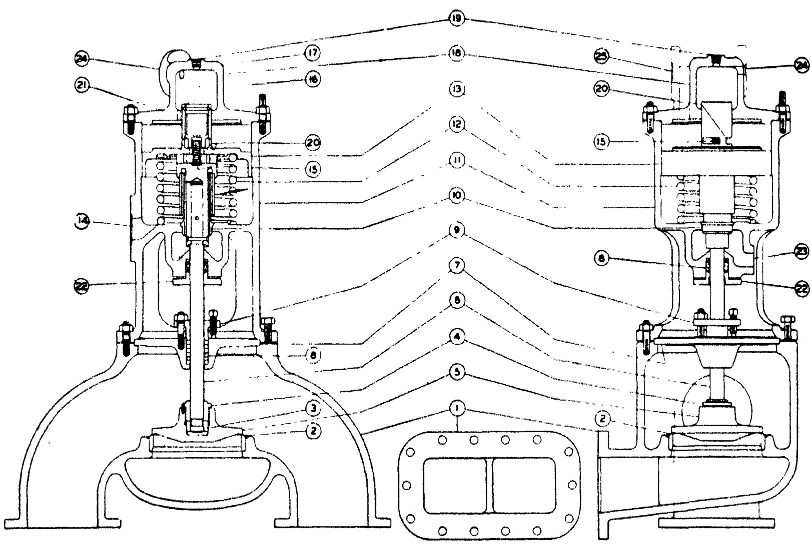
Courtesy of Allis-Chalmers Manufacturing Company

Fig. 9. Gas Turbine Rotor with Blades.

ward, compressing the main spring (12) and closing the by-pass valve (5).

If the emergency governor responds owing to overspeed, the oil pressure vanishes on top of the pilot valve (16). The spring (15) lifts the pilot valve, and the oil on top of the power piston (13) is drained to the oil tank through drains (20) and (14). The unbalanced force from the valve seat and the main spring (12) opens the by-pass valve.

In resetting the plunger of the emergency stop (mounted on bearing housing No. 1), the oil pressure is built up again,



Courtesy of Allis-Chalmers Manufacturing Company

1—Oil Control Housing
2—Valve Seat
3—Spindle Thrust Block Assembly
4—Valve Disc Nut
5—Valve Disc
6—Oil Control Spindle
7—Stuffing Box
8—Packing
9—Stuffing Box Gland (Lower)
10—Piston Sleeve
11—Oil Control Cylinder
12—Piston Spring
13—Oil Control Piston
14—Oil Control Cylinder Main Drain
15—Piston Valve Spring
16—Oil Control Valve
17—Stop Ring
18—Oil Control Cylinder Cover and Dome
19—Oil Pressure Connection for Gauge
20—Piston Release Drain
21—Orifice "A"
22—Stuffing Box Gland (Upper)
23—Auxiliary Drain
24—Oil Supply Line
25—Emergency Drain

Fig. 10. By-pass Safety Valve.

and pilot valve (16) and main valve (5) are closed again.

Provision is made to trip the by-pass valve manually. A pipe, connecting the dome with the drain line, is furnished with a spring-loaded hand-operated valve. Releasing the oil pressure in the dome by means of this valve will open the by-pass valve.

An auxiliary oil drain (23) is provided to catch oil leaking past piston sleeve (10) and to drain it into the main drain system.

Emergency Governor

A centrifugal emergency governor is provided (Fig. 11). This trip (12) operates at 10 per cent overspeed. A branch of the governor oil line (27), which connects to the dome of the by-pass valve, is also connected to the emergency governor. The oil pressure is applied on piston (6) at (28). If the emergency trip (12) operates owing to overspeed, the governor spindle strikes lever (7), which provided the support for the piston (6). The piston is released and is pushed down by the spring (19). Drain slots, provided in the cylinder wall (3) and connecting to the drain connection (16), are uncovered when the piston moves down. The oil pressure in the line to the dome of the by-pass valve vanishes, thereby operating the latter valve in an opening direction. After the speed of the turbine has decreased, the governor spindle of the emergency stop (12) returns to normal position, and the trip piston can be reset by lifting on handle (5). The turbine can be tripped by hand by striking button (10) lightly.

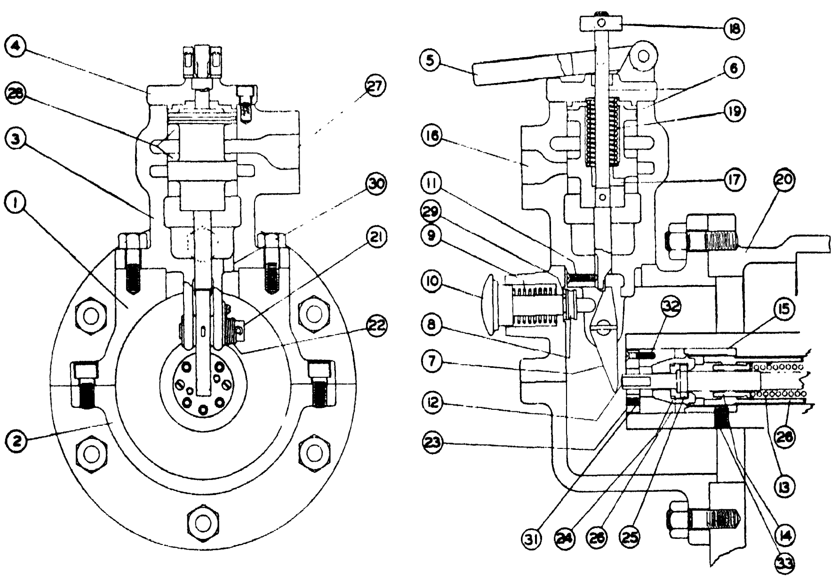
Dismantling, Inspecting, and Reassembling the Governor

To preserve the speed setting of the governor, it should remain on the turbine shaft unless faulty operation requires that it be removed.

In taking off the governor, the first step is to remove the setscrew (33) near the end of the shaft (Fig. 11). Long screws are inserted in holes (31) for pulling the governor out of its bore, but if it has become stuck, it may be necessary to rig up a puller to start the governor from the turbine shaft.

After the governor is out of the shaft, the rivet that locks the governor-spindle lock nut is driven out, and the latter is unscrewed. However, it is well to measure, for future reference, the exact amount the lock nut extends from the governor case (26) before removing the rivet.

The two screws (32) are removed next while holding the governor cover (23) in place. Then the cover is taken off.



Courtesy of Allis-Chalmers Manufacturing Company

1—Governor Housing (Upper Half)
2—Governor Housing (Lower Half)
3—Trip Valve Body
4—Valve Body Cover
5—Resetting Lever
6—Trip Valve Piston and Rod Assembly
7—Trip Latch
8—Lock Washer
9—Spring
10—Hand Trip Pin
11—Setscrew (Guide)
12—Governor Spindle
13—Governor Spring
14—Spring Adjusting Nut
15—Governor Bearing Sleeve
16—Governor Drain Connection
17—Leak Off Drain
18—Collar
19—Spring
20—Bearing Housing
21—Latch Pin
22—Spring
23—Governor Cover
24—Governor Weight
25—Governor Spindle Collar
26—Governor Case
27—Oil Line Connection to By-pass Valve
28—Oil Pressure Chamber
29—Sealing Washer
30—Drain
31—Holes for Pulling Governor
32—Machine Screw
33—Setscrew

Fig. 11. Emergency Governor.

After the cover has been removed and the spring (13) released, the weights (24) may be lifted out with the fingers. The knife edges and seats of the weights are examined.

If the spindle is fouled at the point where it passes through the adjusting nut (14), it will be necessary to remove both for cleaning. To do this, the governor spindle (12) is first removed carefully, the position of the slot in the adjusting nut (14) is marked with respect to the governor case (26), and then the exact distance the adjusting nut (14) projects into the governor case (26) is measured and recorded. Adjusting nut (14) is then screwed out by inserting spindle (12) which is provided with a key which fits in the keyway of the adjusting nut (14). Spindle (12) and adjusting nut (14) are removed and the contact surfaces of each are polished carefully.

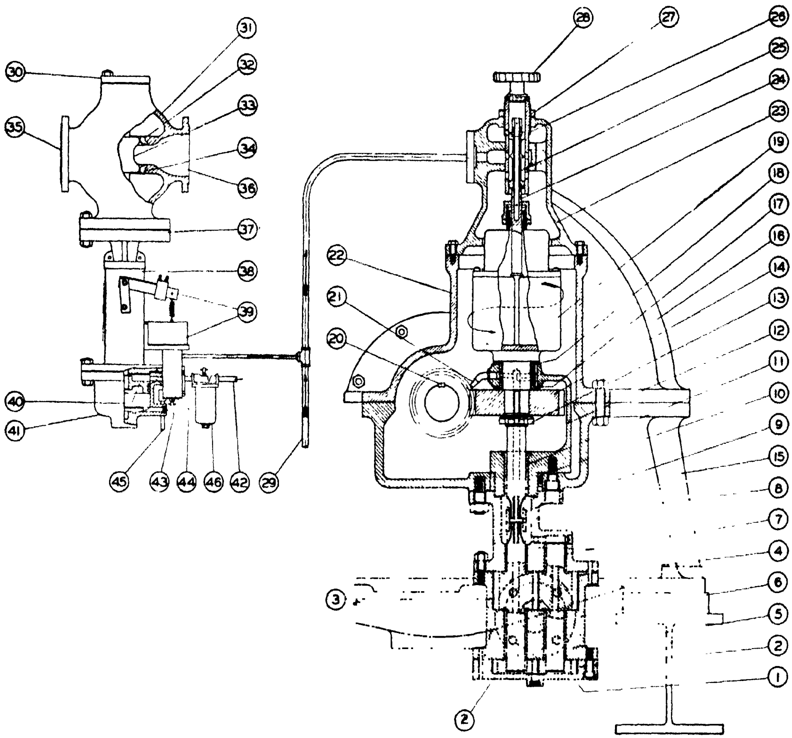
Reassembly is accomplished in the reverse order.

IMPORTANT: No burrs or other slightest obstruction should be raised on the spindle at the point where it passes through the adjusting nut; otherwise, the operation of the governor will be uncertain.

The adjusting nut is screwed back into the case to the exact position it occupied before disassembly. The spindle is inserted and the weights replaced, care being taken to get the knife edges seated properly and to put the spring in place. The governor cover (23) is replaced and the spindle lock nut is screwed on and fastened securely with a rivet. The length of the lock nut extending from the governor case (26) should be checked against the original measurement. The two measurements should correspond because there is only one spot at which the rivet will enter. Deviation between the measurements indicates that the weights are not seated properly.

If the measurements correspond, the governor in the turbine spindle is inserted in such a manner that the holes for the setscrew line up perfectly. The setscrew is put in place, but is backed off ¼ turn after it has bottomed lightly in order to prevent it binding the working parts of the governor. All screws are prick-punched for locking.

After the trip mechanism has been replaced, the turbine is



Courtesy of Allis-Chalmers Manufacturing Company

1—Pump Casing (Bottom Cover)
2—Spur Gear
3—Splined Oil Pump Gear (Driver)
4—Oil Pump Gear (Idler)
5—Bushing
6—Pump Casing
7—Bushing
8—Bearing Case (Oil Pump)
9—Splined Coupling
10—Eccentric Bushing
11—Bushing
12—Governor Shaft Yoke
13—Bearing Lock Nut Assembly
14—Spiral Gear
15—Reduction Gear Casing (Lower Half)
16—Reduction Gear Casing (Upper Half)
17—Thrust Bushing
18—Thrust Bushing
19—Governor Assembly
20—Key
21—Oil Pipe for Spiral Gears
22—Governor Housing and Bearing Cap
23—Pilot Valve Housing
24—Governor Spindle
25—Pilot Valve
26—Pilot Valve Sleeve
27—Lock Nut
28—Hand Wheel
29—Governor Oil Inlet
30—Body Flange Plate
31—Valve Body
32—Inner Seat
33—Stem
34—Outer Seat
35—Connection to Blower Discharge
36—Discharge to Atmosphere
37—Bonnet
38—Cylinder Support
39—Restoring Mechanism
40—Piston Assembly
41—Cylinder
42—Oil Supply to Pilot Valve
43—Connection to Lower Port of Pilot Valve
44—Connection to Upper Port of Pilot Valve
45—Pilot Valve Drain
46—Oil Filter

Fig. 12. Overspeed Protective Mechanism.

brought up carefully to overspeed and the tripping speed of the governor is checked.

Overspeed Protective Mechanism

The overspeed-governor mechanism combines simplicity, ruggedness, and positive control of the turbine with great sensitivity for close speed regulation. The governor mechanism and blowoff valve, shown in Figure 12, provide a protective means against overspeed. The governor is so adjusted as to be inactive at speeds below 3 per cent overspeed. As soon as this amount of overspeed is reached, the blowoff valve begins to open, and air from the blower exhaust is discharged to atmosphere. By means of the hand wheel (28), the speed at which the blowoff valve should open can be adjusted. (This hand wheel is to be kept locked after correct setting.) The governor is so designed that the full opening of the blowoff valve is obtained at 7 per cent overspeed.

The governor mechanism consists of a speed-governor assembly (19), a pilot-valve assembly (25), and an oil-relay-controlled blowoff valve (31) with suitable restoring mechanism to provide an instantaneous response to speed changes.

The speed governor (19) is a standard Allis-Chalmers spring-type governor. The governor weights are secured to two cylindrical rollers rolling on guides provided in the governor casing. These guide surfaces are parallel to the center line of the governor. Flat, flexible springs connect the rollers, the governor casing, and the governor central stem. A coil spring, which is readily adjustable by means of nuts on the spring casing, is provided to load the governor. The governor spindle connects to pilot valve (25), which controls the oil pressure operating the blowoff valve.

As the speed of the turbine increases, the weights which are secured to the rollers tend to move apart owing to centrifugal force. When these forces become greater than the combined forces of the flat and coil springs, the rollers roll upward, and, owing to the fact that the flat spring is attached to the governor spindle, will force the spindle upward.

The spindle motion is transmitted to the pilot valve (25), which moves up and releases the oil pressure in line (29). The resulting motion causes the oil relay on the blowoff valve to move in such a direction as to admit oil pressure from connection (42) through port (44) above piston (40). The piston (40) moves down and forces the oil below piston (40) out through port (43) into the oil relay, where it is drained through connection (45).

The downward motion of piston (40) of the blowoff-valve assembly is transmitted to the restoring mechanism, thus adjusting the opening of the blowoff valve to the desired point.

With any decrease in speed, the reverse action takes place, and in this manner the speed of the unit is effectively controlled. The governor head rotates in sleeve bearings above and below spiral gear (14). The weight of the governor-head assembly is taken on a sleeve-type thrust bearing (17–18) located directly above the spiral gear (14). Although this type of speed governor is extremely sensitive, it is rugged and not subject to variation caused by wear. The only part under stress during operation is the governor spring itself.

Blowoff Valve

A blowoff valve is furnished to bleed off air to atmosphere and to slow down the turbine to normal speed if the unit should happen to speed up. Speed regulation is designed to prevent the unit from reaching the tripping speed (10 per cent overspeed). The valve should be connected in the discharge line of the blower. It is actuated hydraulically by a pilot valve mounted on the speed-reducer gear case. The pilot valve is linked to an overspeed protective mechanism also mounted on the gear case. The valve is normally closed, and starts to open when the overspeed protective governor begins to function at approximately 3 per cent overspeed.

The valve is a standard, flanged 8-inch Smoot valve, normally closed, having a capacity of approximately 810,000 cf per hour with 45 psi (gauge) discharge pressure. The valve body is of semisteel with bronze trim.

Lubrication System

The main oil pump is an Allis-Chalmers gear-type pump driven through gears from the low-speed shaft of the speed reducer (Fig. 13).

The auxiliary oil pump is a motor-driven rotary pump mounted on top of the oil tank.

The oil cooler consists of a tube bundle, suitably baffled and enclosed in a cast-iron shell. The ends of the cooler are provided with removable heads to facilitate inspection and cleaning without disturbing existing pipe connections. Water flows within the tubes, and the oil is directed across the outside of the tubes.

The oil tank is of welded construction, with its cover reinforced to provide a mounting for the auxiliary oil pump. A magnetic, liquid level gauge, float operated, is mounted on the tank. This dial is graduated into *low, mean,* and *high* zones to indicate the oil level in the tank.

The main oil pump takes oil from the tank and delivers it through the cooler to the speed reducer and blower and turbine bearings. A Fisher pressure relief valve maintains about 25 to 30 lb pressure on the bearing supply lines, and by-passes excess oil back to the tank. The bearing drains and gear-case drain are returned to the tank. Straight-through oil-flow sights and thermometers are installed in the drain lines to check the performance of the bearings.

The auxiliary oil pump takes oil from the tank and discharges through a relief valve to the cooler and bearing supply line. This pump also supplies oil to the governor oil system. A pressure gauge checks the auxiliary-pump discharge.

A gauge board is provided for mounting gauges to indicate *bearing-oil, oil-relay,* and *trip-governor* pressures. This board also contains a gauge to indicate the compressor *air-discharge pressure.*

Oil is sprayed on the roller bearings by means of jet pipes which receive oil from the bearing supply line. Each jet pipe contains a small orifice through which a jet of oil is discharged.

The jet strikes a flinger on the shaft which breaks it up into a spray to lubricate the bearing. The gas turbine bearings (Nos. 1 and 2) are equipped with double jet pipes, one on each side of each bearing. The compressor bearings (Nos. 3 and 4) are equipped with one jet pipe on each bearing, each located on the coupling side of the bearing.

On some compressors, where a double roller bearing is used, the oil is introduced through a passage drilled through the bearing retainer. This passage is located between the two bearings and is fitted with a metering orifice. Oil enters the bearing through this orifice and flows outward in both directions between the inner and outer races of the bearings.

Oil is supplied to the Kingsbury thrust bearing (No. 5) through two inlet pipes near the bottom of the bearing. Each inlet pipe contains a flanged connection located between the main bearing oil supply line and the compressor casing. Each of these connections is fitted with a metering orifice plug and a dial gauge, which indicates the pressure of the oil on the bearing side of the orifice.

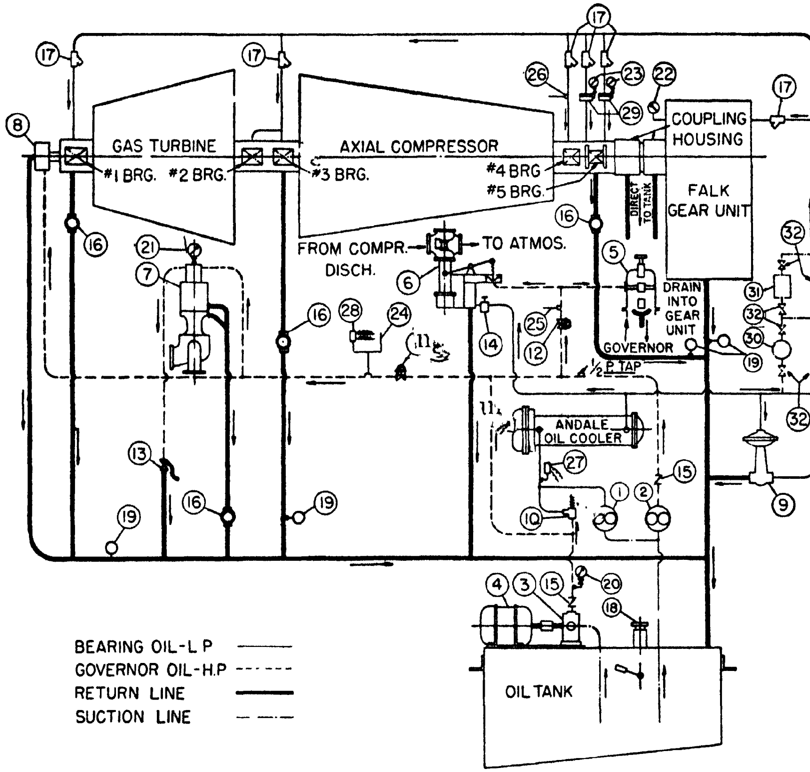
The coupling between the compressor and the speed reducer is lubricated by means of a jet pipe which delivers oil to the oil-collector rings on the coupling. This jet pipe is located on the compressor side of the coupling.

Oil for lubricating the speed reducer is taken from the bearing supply line.

Governor Oil System

The high pressure oil required for the governor oil relay, by-pass safety valve, and overspeed trip valve is supplied by a second gear-type pump driven through gears from the low-speed shaft of the gear reducer, Figure 13.

The pump takes its suction from the oil tank and delivers oil through one adjustable needle valve to the by-pass valve and overspeed trip, and through another needle valve to the governor oil relay. The drains from the by-pass valve and the emergency overspeed trip valve are brought back to the oil tank. A quick-opening valve in a by-pass line from the dome to the tank is provided for emergency manual tripping of the



Courtesy of Allis-Chalmers Manufacturing Company

1—Main Oil Pump—L. P.
2—Gov. Oil Pump—H. P.
3—Aux. Oil Pump
4—Aux. Oil Pump Motor
5—Gov. Oil Pilot Valve
6—"Smoot"-Cyl. Oper. Blow-Off Valve (Compr.)
7—By-pass Safety Valve (Gas Turb.)
8—Emergency Gov. Trip Valve
9—"Fisher" Reducing Valve
10—Relief Valve
11—York Capped Needle Valve (Trip Gov.)
12—York Capped Needle Valve (Oil Relay)
13—Quick Opening Valve
14—Oil Filter
15—Check Valve
16—Oil Flow Sights
17—Oil Strainer
18—Oil Tank Liquid Level Gauge
19—Weston Dial Thermometer
20—Oil Press. Gauge—Aux. Oil Pump
21—Oil Press. Gauge—By-pass Safety Valve
22—Oil Press. Gauge—Gear Unit
23—Oil Press. Gauge—Thrust Bearing
24—Oil Press. Gauge Conn. (Trip Gov.) Mounted on Special Gauge Board
25—Oil Press. Gauge Conn. (Oil Relay) Mounted on Special Gauge Board
26—Oil Press. Gauge Conn. (Brg. Oil) Mounted on Special Gauge Board
27—Mercoid Press. Switch—0-35# with Explosion Proof Cases Mounted by Customer
28—Mercoid Press. Switch—0-100# with Explosion Proof Cases Mounted by Customer
29—Orifice Plug for Thrust Bearing Oil
30—Nugent Oil Filter (Furnished & Installed by Customer)
31—Blotter Pressure Filter (Furnished & Installed by Customer)
32—Globe Valves (Furnished & Installed by Customer)

Fig. 13. Schematic Oil-piping Arrangement.

by-pass safety valve. Oil supplied to the power piston used to operate the blowoff valve is taken from the bearing oil supply line. Pressure in this line is maintained by a reducing valve which by-passes excess oil to the drain.

Auxiliary Oil-pump Control

The auxiliary oil pump is provided to supply oil to both the governor oil system and the low pressure system (bearings) during starting of the gas turbine unit, and also to serve as a stand-by unit for the main governor oil pump and low pressure pump (Fig. 15).

For starting the gas turbine unit and bringing it up to speed, the auxiliary oil pump is started by pushing the *start* button on the push-button station provided for this purpose. After the gas turbine has come up to speed, the auxiliary pump can be stopped by pushing the *stop* button.

The auxiliary oil pump also serves as a stand-by for the main governor oil pump and the low pressure pump. In both the governor oil line and the low pressure line are pressure-operated *mercoid* switches (Type DA–31–127) which are adjusted to close and start the auxiliary oil pump when the pressure falls slightly below normal. A pilot light on the auxiliary pump push-button station immediately informs the operator that the auxiliary pump is running. If he pushes the *stop* button and finds that the auxiliary pump continues to run, he will be immediately informed of a failure in one of the main pumps. If this proves to be the case, the cause of failure should be determined and repaired at the earliest possible moment.

Erection—General Instructions

The unit is to be erected according to foundation and general arrangement drawings furnished by Allis-Chalmers. A suitable foundation and subfoundation to suit local conditions must be built by the user.

Turbine, compressor, and gear are mounted on a single bed plate. The turbine and compressor spindles are joined by a rigid coupling and are supported by four roller bearings. The outer race of each bearing is mounted in a spring ring which

is provided to aid in alignment and to allow for radial expansion of the bearing. Beginning from the outboard end of the turbine, the bearings are numbered 1, 2, 3, and 4 respectively. No. 1 bearing is mounted on the outboard end of the turbine cylinder. No. 2 and No. 3 bearings are mounted on the compressor cylinder, one on each side of the coupling. No. 4 bearing is mounted on the compressor cylinder at the outboard end. A Kingsbury thrust bearing, located adjacent to No. 4 bearing, maintains the axial position of the two spindles and takes any thrust which may be encountered.

The turbine and compressor cylinders are each supported at their four corners on keys which lie in transverse keyways cut in the bed plate. The cylinders ends lying adjacents to one another are keyed to the bed plate in a manner that prevents any axial movement of the cylinders at that point. The opposite ends of the cylinders rest on keys, but are free to slide, thereby taking care of any expansion caused by heating. Transverse movement of the cylinders is prevented by cylinder guides fastened to the bed plate along the center line of the unit. Two guides are provided for each cylinder, and on some units, an additional guide is provided for the No. 1 bearing pedestal. The final positions of these guides are to be determined in the field. After the unit has been aligned, the guides are to be fitted to their keyways in the cylinders by means of the gib keys furnished with the machine.

When erecting the unit, the leveling pads in the foundation should be checked to see that they lie in a level plane. The bed plate is placed upon its foundation. Then the lower half of each cylinder is set on the bed plate and the bearing bores are lined up. When lowering the spindles into place, the lifting rigs provided should be used. The spindle should be checked to see if it hangs level in the lifting rig, care being taken not to damage the labyrinth sealing strips. The guide pins (if any) in the bearing spring rings should be pointing vertically upward when the bearings are in place.

The unit was carefully aligned and given a test run at the factory. However, in the field it will be necessary to recheck the alignment and to shim up the bed plate, if necessary, to

again obtain the proper alignment. To check the alignment, the faces of the rigid coupling are separated slightly, and by means of a thickness gauge, the two faces are checked for parallelism. After these faces have been made parallel, a dial indicator is attached to the compressor half of the coupling with the indicator following the hub of the turbine half of the coupling. The compressor spindle is rotated to check whether the centers of the spindles are in line. By using a small mirror in the bottom of the coupling housing, the dial can be read regardless of its position.

After the first alignment is obtained, the foundation bolts should be pulled up fairly tight, and the alignment rechecked. When the alignment is completed, the unit may be grouted in. A two-to-one mixture of sand and cement, mixed fairly soft, should be used. It is desirable to completely fill the bed plate with grout where possible. All necessary foundation bolts, nuts, washers, shims, and wedges are furnished by the user.

The Kingsbury thrust bearing and its housing may be assembled to the compressor shaft before the spindle is lowered into the cylinder. The bearing should be assembled with the filler piece located at the outboard end. The thrustbearing housing has a tongue which fits between two lock rings in a groove cut in the lower half of the compressor cylinder. These lock rings, stamped *X* and *Y*, were fitted at the factory and hold the housing in its correct position. Corresponding letters stamped in the compressor cylinder indicate the correct positions of the lock rings. When the housing is in place, the rings should be inserted from the side of the housing stamped to correspond to the ring.

In the plane where the faces of the rigid coupling meet, a line has been scribed on the surface of the compressor cylinder as a reference to be used when checking the radial clearances of the turbine blading. It can also be used as a guide during erection until the thrustbearing housing is locked in place.

Clearances

Radial clearance of all compressor blades should be equal to or slightly more than 0.035 in. Experience has shown that

the compressor has ample clearance when a 0.035-in. thick shim can be pushed through each row of blading between the tips of the spindle blading and the lower half of the cylinder, and between the tips of the cylinder blading and the spindle. Clearance for the upper half of the cylinder can be obtained by the lead-wire method.

Radial clearances for all turbine blading can be obtained by the lead-wire method. When taking these clearances, the face of the coupling is brought up to the line scribed on the surface of the compressor cylinder. This position can be checked by laying a straightedge on the line. It would be well for the user to make and keep a record of the original clearances of the turbine for future use.

Piping

Piping to and from the compressor and turbine *must* be properly supported to prevent weight strains from being transmitted to the compressor or turbine casings. Proper allowance must be made for the expansion of the compressor and turbine to avoid distortions of the casings. Expansion joints are recommended for both inlet and exhaust lines of the turbine, and for inlet and discharge lines of the compressor.

Erection—Precautions

During erection, the following points should be carefully checked and necessary adjustments made before attempting to start the unit:

1. Check alignment carefully after the air and gas piping have been installed.

2. Thoroughly clean all bearing housings and all parts of the lubrication system before installing. All pipes have been pickled. Parts should be flushed with kerosene and blown out with an air blast. Do *not* use waste cloth to wipe parts dry.

3. For flushing the system, fill the oil tank with a light oil. Check the direction of rotation and operation of the auxiliary oil pump. Circulate the oil through the system. When the pressure drop across the filter starts to build up, turn the filter hand wheel once or twice to clean the filter. Continue circu-

lating oil until the system has been thoroughly flushed; then drain the oil and clean the tank.

4. Refill the oil tank with a good grade of clean turbine-quality lubricating oil, which is suitable for high-speed bearings service. The oil should have a viscosity of 350 sec Saybolt Universal at 100° F.

5. Check the oil and cooling water systems for leaks. Adjust the lubrication and governing systems.

6. Adjust reducing valve (10) in the auxiliary oil-pump line to give about 60 lb pressure at the auxiliary-pump discharge (Fig. 13).

7. Adjust the Fisher reducing valve (9) to maintain about 30 lb pressure on the bearing oil pressure gauge (on the gauge board).

8. Adjust needle valve (12) temporarily to give 40 to 60 lb pressure on the governor relay system. After the main unit is brought up to speed, further adjustment of this valve is necessary.

9. Open needle valve (11), which is intended to limit the flow of oil through the trip valve after the emergency trip has operated. After the main unit is running, make further adjustments of valve (12) so that approximately 40 to 60 lb pressure is maintained on the oil relay pressure gauge; then turn down needle valve (11) until the pressure, shown on gauge (21) located on the by-pass dome, just begins to fall.

10. Make electrical connections to mercoids. Set mercoid pressure switch (27) to cut in at approximately 18 lb pressure. Set mercoid pressure switch (28) to cut in at approximately 50 lb pressure.

11. Operate quick-opening valve (13) and see that the piston controlling the by-pass valve operates freely. The total travel of the piston is shown by the pointer fastened to the valve stem.

12. Trip the overspeed trip valve (8) manually to see if it operates freely.

13. (a) If a starting motor and magnetic clutch are used for starting, check the direction of rotation of the starting motor and the operation of the clutch. The starting-motor

bearings should be filled with a suitable oil or grease according to the manufacturer's instructions. Full instructions for the clutch line-up should be obtained from the clutch manufacturer. (b) A starting turbine, if used, should be installed according to the manufacturer's instructions.

14. Phase out generator leads and mark them to correspond with line leads.

15. Adjust Smoot blowoff-valve mechanism according to the manufacturer's instructions.

16. Roll the unit slowly with the starting unit, and check carefully to see that everything is free.

17. After satisfactory mechanical operation has been obtained, the turbine and compressor horizontal joints should be made permanent. Use a mixture of triple-boiled linseed oil and graphite of a consistency such as will run off the brush with which it is being applied.

Starting (for Steam-Turbine-Driven Unit)

1. Start up the auxiliary oil pump.
2. Check oil pressures and oil sight feeds.
3. Check the oil in the starting turbine for correct level.
4. Test the gas turbine constant-speed governor by screwing the hand wheel of the main governor linkage all the way down and then all the way up. Leave the wheel all the way *down.*
5. Test the gas turbine overspeed dump valve by tripping the overspeed latch. Reset.
6. Test the steam turbine overspeed trip. Reset the trip.
7. Open the cooling water to the steam turbine bearings.
8. Open all steam turbine drain valves.
9. Open the steam turbine exhaust valve.
10. Crack the steam turbine throttle valve and bring it up to part speed. Operating instructions furnished by turbine manufacturer should be carefully followed. Immediately check to see that the oil rings are revolving.
11. Start the cooling water when the return oil temperature from the bearings reaches 125° F. Regulate the flow to maintain 125 to 130° F.

12. Check the bearings for sound and temperature. Check the glands for sound. Open the valves to the turbine gland seals. Check the sound of the rotors for rubbing.

13. When the water is out of the steam system, close all the drain valves.

14. Open the steam throttle to bring the unit up to approximately 40 per cent speed.

15. Check again as in Step 12.

16. Light the burners, if necessary, to bring the inlet gas temperature to 850° F.

17. Cut down on the steam throttle as the gas turbine comes in under its own power.

18. Shut down the auxiliary oil pump manually. If the governor oil pump and main oil pump do not maintain the operating pressures, the auxiliary pump will continue to run. This condition should be corrected before putting the unit in service.

19. Shut down the steam throttle valve.

20. Keep the steam turbine exhaust valve open.

21. Open all steam turbine drains.

22. Regulate the gas turbine speed to rated speed with the hand wheel on main governor. Run the inlet gas to the turbine to 850° F. Maximum continuous allowable temperature is 950° F. Shut down the burners, if possible, when the unit is on the Houdry system.

23. Start an operations log. Pay close attention to the temperature of the oil discharged from the bearings. In case of abnormal bearing temperatures, shut down the unit, determine the cause of overheating, and make corrections before again placing the unit into operation.

Operation

The compressor is essentially a constant-volume machine and delivers a steady flow of air when operating in the stable zone. If the discharge pressure is raised sufficiently by increasing the resistance in the discharge line, the compressor will begin to *pump*—that is, puffs of air will be blown back out of the compressor inlet. This condition should be avoided. Pump-

ing can be prevented by use of an antipumping regulator arranged to increase the volume of air handled by blowing off air at the compressor discharge when the pumping zone is reached.

The unit will run with a minimum of vibration at all speeds, except when passing through the critical speed zone.

In case of excessive vibration, the unit should be shut down, and the cause determined and corrected at once.

Excessive vibration may be caused by:

1. Misalignment.
2. Rubbing of rotor or stationary blades caused by:
   (a) Distortion of casing.
   (b) Excessive heating of unit.
3. Defective bearing.
4. Defective gears.

The unit should be brought up to speed slowly, and allowed to cool slowly when being shut down.

The alignment should be checked frequently when the unit is first placed into operation.

The oil cooling water must be clean and free from dirt, and supplied at a steady pressure. About 100 gpm of 85° F water should be sufficient. Water pressure drop across the oil cooler will then be approximately two pounds.

The bearing-oil pressure should be approximately 20 to 30 lb (gauge). The oil discharged from the bearings should be maintained below 150° F.

The relay-oil pressure should be approximately 40 to 60 lb (gauge).

The governor-oil pressure should be approximately 60 to 95 lb (gauge). Inspection of the pressure gauges and thermometers should be a part of the operator's daily routine.

Stopping (Normal Shutdown)

1. Reduce the generator load to zero, and take generator off the line (for units provided with a generator).
2. Put out the burners if they are operating.
3. Start the auxiliary oil pump when the oil pressure be-

gins to decrease. Check the auxiliary-pump discharge pressure. *Note:* If the auxiliary pump is equipped with automatic control, the pump should start up automatically in case of low oil pressure.

4. (a) *For starting motor:* When the unit has slowed down sufficiently, start up the starting motor and engage the clutch.

(b) *For starting turbine:* Crack the throttle valve sufficiently to keep the unit rotating at slow speed.

5. Keep the unit turning over until the turbine-cylinder temperature has dropped to 250 to 300° F.

6. Shut down the driving motor or turbine. For turbine drive, open the drains, and so forth, according to the manufacturer's instructions.

7. Shut off the cooling water.

8. Shut down the auxiliary oil pump.

Stopping (Emergency Shutdown)

1. Trip the overspeed lever, located at the outboard end of the turbine.

2. Start the auxiliary oil pump.

3. Proceed as outlined above if the unit can be kept running. If the machine cannot be kept rotating, let the auxiliary oil pump run, and turn the rotor by hand—½ turn every 15 minutes.

Maintenance

The upper-half casings of the compressor, turbine, and gear should be removed at regular intervals, usually one year, for inspection of the rotating parts. Services of a capable erection engineer may be obtained by applying to Allis-Chalmers.

The bearings should be inspected at frequent intervals. Access to the bearings is obtained by removing the bearing covers.

The lubricating oil should be periodically filtered, and water and sludge in the oil reservoir drawn off frequently. The oil should be renewed if its appearance or excessive sludging indicates decomposition has set in.

The filters in the oil lines should be cleaned regularly.

The tube bundle should be removed from the oil-cooler shell and cleaned at intervals.

Dismantling and Repairs

The compressor and turbine upper-half casings may be lifted after removing the connecting piping and bolts. The casing should be lifted slowly and evenly to avoid binding on the cylinder guides.

To remove the compressor or turbine rotor:

1. Remove the bearing covers.
2. Disconnect the couplings.
3. Lift the rotor, using the lifting rig furnished with the unit. Separate rigs are provided for the turbine and compressor rotors.

Note: The coupling between the blower and turbine has a male and female fit. The turbine rotor *must* be moved about 3/16 in. axially to clear this shoulder before lifting the rotors.

The rotors were statically and dynamically balanced, and given a mechanical test before shipment. Balance was obtained by adjustment of balance screws located at each end of the rotor. Should the rotor become unbalanced for any reason, Allis-Chalmers should be consulted.

Chapter IV

Operating Instructions for Neuchatel 4000-Kw Plant[1]

This chapter answers many questions concerning the way in which a large central station gas turbine unit is operated, how it is brought up to speed and load, and how it is shut down. Particularly worthy of note is that a 4000-kw gas turbine unit is not as complicated as a steam plant of the same size. However, such a gas turbine unit cannot be brought up to speed as quickly as a jet unit.

The combustion turbine may be started in two ways: (1) nonautomatically and (2) automatically, in case of current failure in the system or after having been switched off by push button.

In both cases, the station must be in the same state of preparedness except for the position of the change-over switch marked *nonautomatic start/automatic start.*

In order to assure effective starting, operation, and availability of the station at all times, the following prescriptions should be observed very carefully.

Nonautomatic Starting

1. The change-over switch is in position *nonautomatic start.*
2. Switch the system change-over switch to *Diesel.*
3. Start the Diesel engine.
4. (a) Start the auxiliary oil pump. Check the pressure of the pump on the gauges, the oil circulation in the

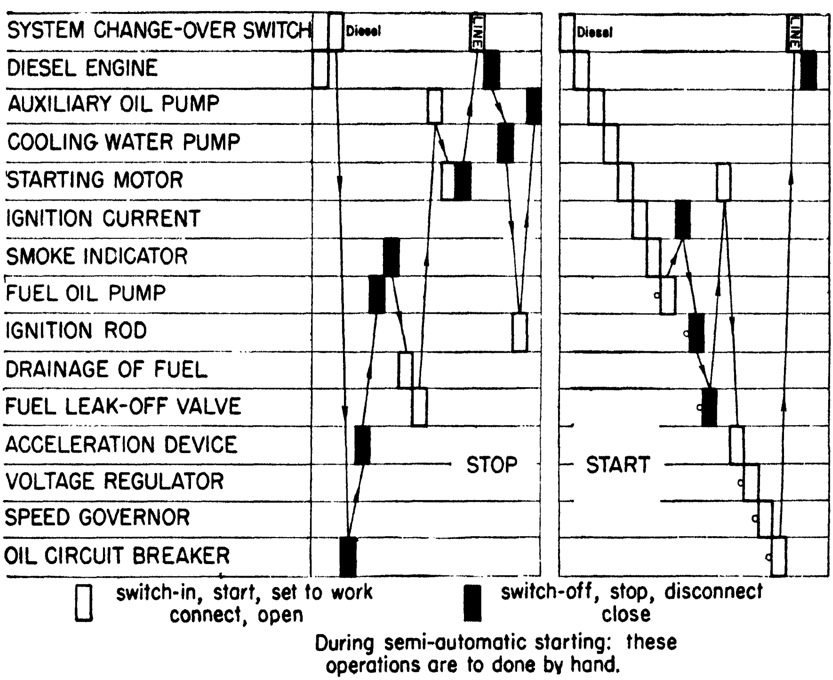
[1] Information furnished by Paul R. Sidler, President, Brown Boveri Corporation, New York

bearings, and be sure that the by-pass of the combustion turbine is closed.

(b) Start the cooling water pump.

5. Switch in the starting motor. The time interval from one step of the rotor starter to the next should be at least 10 sec. If this is not the case, regulate relay 191.[2] Check the operation of the auxiliary fan.

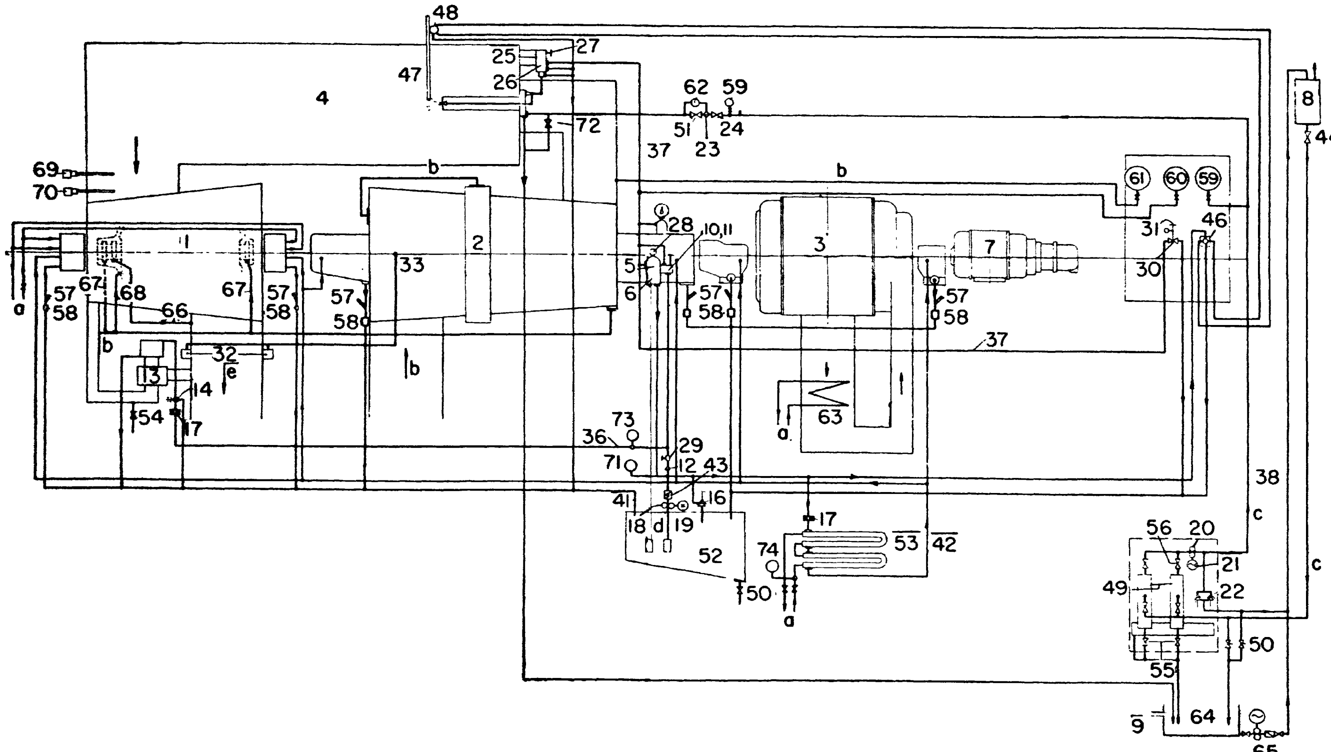
6. At about 600 rpm, switch in the ignition current and check its rating, which should be 23 amp.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 1. Schedule of Operation.

[2] Relay 191 connects and disconnects the motor control of the rotor starter. The motor control moves the starter from Step zero to Step 10, and then stops the starter. This corresponds to a speed of the starting motor or the gas turbine of 750 rpm, at which speed the combustion is ignited. After ignition, a push button restores the connection of the motor control, and the starter advances from Step 10 to 16, causing the starting motor to reach its maximum speed of about 1000 rpm. At this speed, the starting motor is disconnected electrically by a centrifugal switch, and its brushes are lifted. However, the switch remains coupled mechanically to the gas turbine so that the latter's rotor will also reach the normal turbine speed of 3000 rpm.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 2. Diagram of Pipe Connections.

List of Parts on Opposite Page

7. At Step 10 of the rotor starter (when load on starting motor begins to fall off):

(a) Switch in the smoke indicator.

(b) Switch in the fuel-oil pump and check the ignition. If the fuel does not ignite, stop this pump and investigate the causes (see 10(a) below).

KEY TO FIG. 2, PAGE 62

1—Combustion Turbine
2—Compressor
3—Generator
4—Combustion Chamber
5—Speed Governor
6—Gear Oil Pump
7—Starting Motor
8—Fuel Tank
9—Overflow
10—Safety Regulator
11—Starting and Stopping Device
12—Oil Filter
13—Safety Valve
14—Pressure-adjusting Valve
16—Pressure-adjusting Valve
17—Adjusting Diaphragm
18—Auxiliary Oil Pump
19—Driving Motor for 18
20—Fuel Oil Pump
21—Driving Motor for 20
22—Pressure-Adjusting Valve for Fuel Oil
23—Fuel-oil Filter
24—Fuel-oil Valve
25—Fuel-oil Regulator
26—Piston of 25
27—Hand Regulation for 25
28—Tachometer
29—Regulating (Adjusting) Screw
30—Acceleration Device for Starting Motor
31—Hand Regulation for 30
32—Smoke Indicator
33—Air for Smoke Indicator
36—Safety-Regulator Oil System
37—Governor Oil System
38—Fuel-oil Piping
40—Air Pipe
41—Oil-return Pipe
42—Lubricating Oil
43—Nonreturn Valve
44—Stop Valve
46—4-way Valve
47—Ignition Rod
48—Control of Ignition Rod
49—Fuel-oil Filter
50—Drain Valve
51—Stop Valve
52—Lubricating Oil Tank
53—Oil Cooler
54—Drain Valve
55—Drain Valve for Fuel Oil
56—Stop Valves for Filter 49
57—Thermometer
58—Peephole to Check Oil Circulation
59—Pressure Gauge for Fuel Oil
60—Pressure Gauge for Governing Oil
61—Pressure Gauge for Air After Compressor
62—Fuel-oil Meter
63—Generator Air Cooler
64—Leakoff Tank (Sump)
65—Leakoff (Sump) Pump
66—Gas Drain Pipe from Stuffing Box
67—Sealing Air
68—Cooling Air
69—Thermostat for Speed Governor
70—Thermostat for Acceleration Device
71—Pressure Gauge for Safety Oil System
72—Drain Valve for Fuel Oil
73—Pressure Gauge for Oil after Pump
74—Pressure Gauge with Signalling Device for Cooling Water.

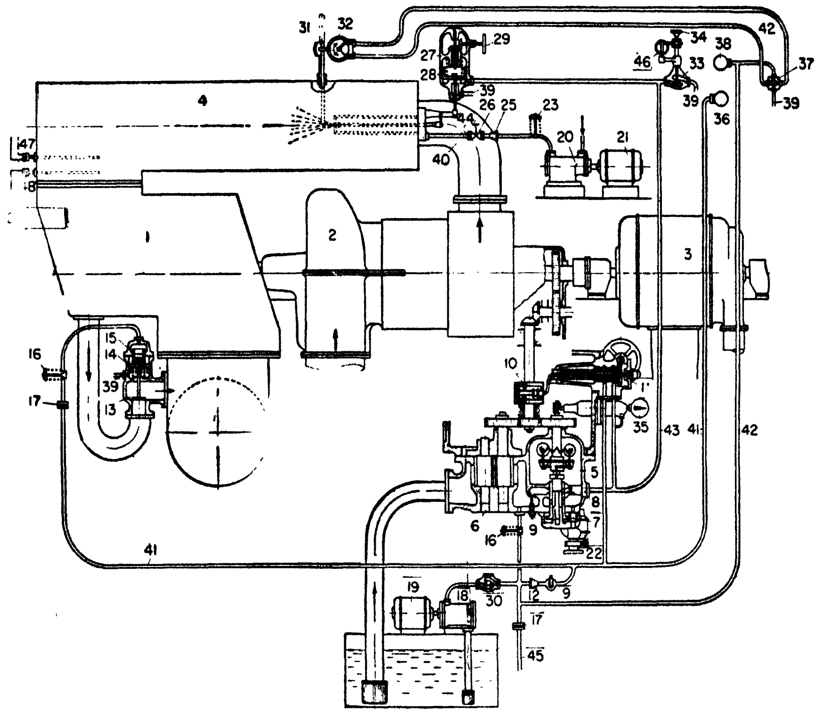
a—Cooling water
b—Sealing air
c—Fuel-oil circuit
d—Lubricating and governing oil circuit
e—Gas exhaust

(c) When ignition is completed:
(1) Switch out the ignition current.
(2) Retract the ignition rod.
(3) Close the leakoff (return valve) in the fuel line.
(4) Accelerate the starting motor; the rotor starter advances from Step 11 to 16.

8. Operate the starting device by hand or by motor with a view to avoid creating smoke and to prevent too rapid acceleration of the machine. Stop in a given position or even return to the preceding step. *The acceleration up to 3000 rpm should normally be completed in 10 min.*

9. (a) If the set is to work on the system alone:
(1) Regulate the voltage to 3900 v.
(2) Regulate speed to obtain 50 cycles.
(3) Switch in the oil circuit breaker.
(4) Change the system change-over switch to position *network.*
(5) Stop the Diesel engine.
(6) Put on load; keep normal speed.
(7) Bring the Diesel engine back into starting position.

(b) If the set is to work in parallel:
(1) Regulate the voltage to conform to that of the network.
(2) Vary the speed to conform to the network frequency (synchroscope).
(3) *Do not* switch in the oil circuit breaker unless the phase voltmeter stops for a moment between zero and 20 v.
(4) Change system change-over switch to position *network;* check the ammeters of the fuel-oil and cooling-water pump sets.
(5) Stop the Diesel engine.
(6) Put on load; keep normal speed, adjust the power factor.
(7) Bring the Diesel engine back into starting position.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

1—Combustion Turbine
2—Compressor
3—Generator
4—Combustion Chamber
5—Main Speed Governor
6—Gear Oil Pump
7—Regulating Sleeve
8—Overflow—Part for Sleeve 7.
9—Adjusting Screw
10—Safety Regulator
11—Shut-down (Overspeed) Device
12—Oil Filter
13—Safety (By-pass) Valve of Turbine
14—Operating Piston for 13.
15—Control Piston for 13
16—Safety Valve for Oil Circuit
17—Diaphragm
18—Auxiliary Oil Pump
19—Driving Motor for 18
20—Fuel-oil Pump
21—Driving Motor for 20
22—Speed-adjusting Device
23—Safety Valve for Fuel
24—Fuse for 23.
25—Fuel-oil Filter
26—Stop Valve
27—Servo-motor of Burner
28—Operating Piston for 27
29—Hand Regulation for 27
30—Nonreturn Valve
31—Ignition Rod
32—Servo-motor for 31
33—Control for Rotor Starter
34—Hand Regulation for 33
35—Tachometer
36—Pressure Gauge for Oil Circuit 41
37—4-way Valve
38—Pressure Gauge for Oil Circuit 42
39—Oil-return Pipes
40—Air Pipe
41—Oil Circuit for Safety Governor
42—Oil Circuit for controlling 32
43—Oil Circuit for Speed Governor
44—Fuel-oil Pressure Pipe
45—Lubricating-oil Piping
46—Electric Motor for 33
47—Thermostat for 46
48—Thermostat for 22

Fig. 3. Governing Diagram.

10. Difficulties during starting:
   (a) If the fuel does not ignite:
      (1) Check if the leakoff valve in the fuel line is not too widely open: one-quarter turn from closed position is normal.
      (2) Check the incandescence of the ignition rod, which should be cherry red.
      (3) Check the fuel-oil pressure (17 kg per sq cm), and its atomization in the combustion chamber.
      (4) If points (1)-(3) are in order, it is still possible that the burner is clogged; clean the burner according to instructions or replace it by a spare.
   (b) The change-over switch is in position *automatic start.*
   (c) Either by the failure of the current in the system or by the push button, *test starting,* the machine is automatically started as follows:
      (1) Change the system change-over switch to *Diesel.*
      (2) Start the Diesel engine.
      (3) Start the lubricating oil and cooling water pumps.
      (4) Switch in the starting motor and accelerate to 750 rpm (in about 100 sec).
      (5) Switch in the ignition current.
      When 750 rpm are reached (Step 10 of rotor starter) and the ignition current is on, the machine waits for:
   (d) The first intervention of attendant:
      Switch in the fuel-oil pump.
      When ignition is completed, retract the ignition rod. This releases the following automatic operations:
      (6) Switch off ignition current.
      (7) Advance the rotor starter.
      (8) Acceleration device operates to bring the machine to 3000 rpm.

Close the leakoff (return valve) in fuel line.
When 3000 rpm are reached, the machine waits for:

(e) The second intervention of attendant:
Switch in the oil circuit breaker or synchronize the machine. *Watch for the stop of the phase voltmeter at zero voltage.* This causes the following automatic operations:

(9) Change the system change-over switch to position *network*

(10) Stop the Diesel engine. *The automatic starting is now complete.*

(f) Return the change-over switch to position *non-automatic start* to avoid undesired starting operations.

(1) Put on load; keep normal speed, adjust the power factor.

(2) Bring the Diesel engine back into starting position.

Normal Operation, Checks, and Instrument Readings

The following characteristics should be kept constant:

Lubricating and Governing Oil

Delivery pressure of pump ......4.5 kg per sq cm

Lubricating oil pressure ....1 kg per sp cm

When this oil is cold, the pressure is 0.3 to 0.5 kg per sq cm higher.

Temperature after lubricating oil cooler ....40 to 50° C

Temperature after bearings ....50 to 60° C

Combustion Chamber

Pressure of fuel oil ....17 kg per sq cm

Turbine

Maximum temperature at inlet ....570° C

Speed ....3000 rpm

Generator

Voltage ....3900 v

The temperature of the lubricating oil may be reduced by increasing the cooling water quantity, that is, by opening the water valve into the oil cooler, and vice versa. The attendant will check the lubricating system from time to time. The operation of the cooling water pump set should be checked regularly. (The normal current input is 8 amp.)

Stopping the Set

The set is always stopped by hand operation (nonautomatic) (Fig. 1).

1. Start the Diesel engine.
2. Change the system change-over switch to position *Diesel.* Check the ammeters of the fuel-oil and cooling water pumps.
3. Remove load to zero; then open oil circuit breaker.
4. Open completely the acceleration device.
5. As soon as the speed no longer drops appreciably (at about 2000 rpm):
   (a) Switch off the fuel-oil pump.
   (b) Switch off the smoke indicator.
   (c) Open the drain valve of the fuel-oil system.
   (d) Open completely the leakoff valve of the fuel pipe.
6. At about 1000 rpm, switch in the auxiliary oil pump.
7. After having placed in operating position the brushes of the starting motor (at about 700 rpm):
   (a) Switch in the starting motor, which will keep the speed of the set at about 750 rpm. Check Step 10 of the rotor starter.
   (b) Stop the starting motor, when the air temperature at the turbine inlet has reached 60 to 70° C. Check to see if rotor starter is returned to zero (starting) position.
   (c) If there is current in the network, change the system change-over switch to position *network* and check the ammeters of the auxiliary oil pump and the cooling water pump.
   (d) Stop the Diesel engine, and return it to starting position.

8. At about 100 rpm:
   (a) Disconnect the cooling water pump and close the valves.
   (b) Insert the ignition rod into starting position.

9. When the machine comes to a standstill disconnect the auxiliary oil pump.

Chapter V

Gas Turbine Locomotive Construction[1]

A brief description and test data of the gas turbine electric locomotive of 2200 hp built by Brown Boveri & Co., Ltd. for the Swiss Federal Railways was given in the book *The Modern Gas Turbine.* The object of this chapter is to give the detailed construction of a later design that is especially suited for American railroad conditions.

The operating crews appreciate the simplicity of operation and the flexibility of the power plant. In view of this record, the locomotive has meanwhile been definitely accepted by the Swiss Federal Railways and is now in the service together with other locomotives powered by internal-combustion machines.

The gas turbine power plant of the Swiss locomotive has already incorporated all the experience previously gained with gas turbines in other applications. In the electrical equipment and controls, Brown Boveri could look back upon several decades of consistent and successful work in the electric and Diesel-electric locomotive fields.

A year of regular service confirmed the results of the earlier trial runs in regard to guaranteed performance. All requirements have been fully met, no major disturbances of any kind have occurred. Minor adjustments could always be completed during scheduled waiting periods. Therefore, the availability record is excellent.

Based on this background and using the data collected during the operation of the gas turbine locomotive since the fall of 1941, Brown Boveri has now developed a *standard* gas turbine electric power plant for locomotive use. In its main ele-

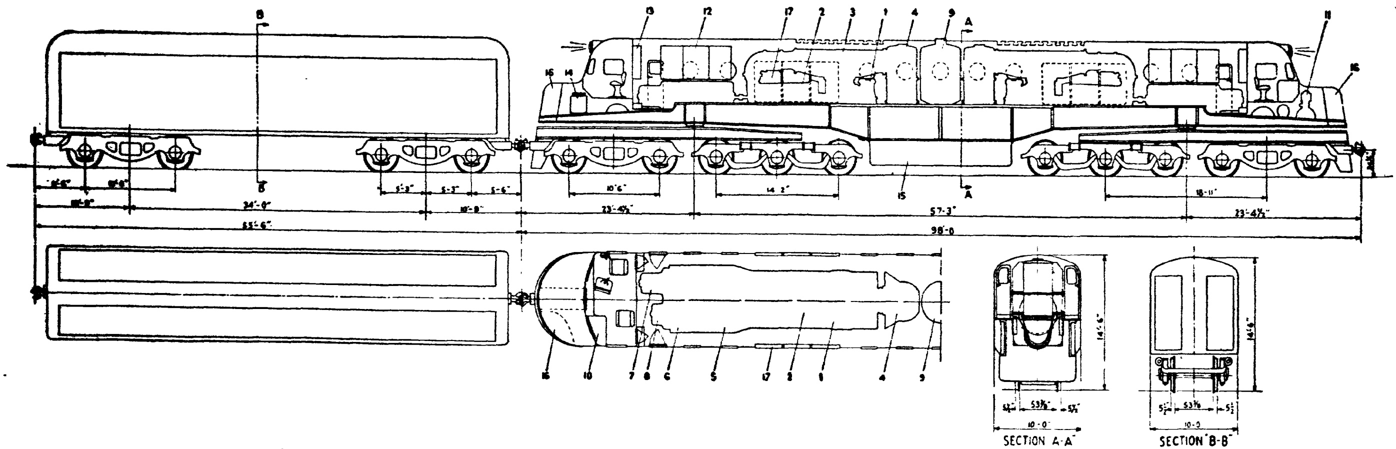
[1] Revision of a paper presented before the A.S.M.E. by Paul R. Sidler, President, Brown Boveri Corporation, New York.

ments, this locomotive uses the same frame sizes as the Swiss locomotive, but incorporates several improvements in design details which result in a net output of 2500 hp and a thermal

TABLE 1

GENERAL DATA OF GAS TURBINE LOCOMOTIVE USED IN BOTH PASSENGER AND FREIGHT SERVICE

Item	5,000 Hp. Single Unit and Tender	2,500 Hp. Front Unit	5 000 Hp Front and Center Unit	5,000 Hp. Both end Units	7,500 Hp All Three Units
Outline drawing	Fig 1	Fig. 3	Fig. 2	Fig. 2	Fig 2
Gage of track ft-in	4–8½	4–8½	4–8½	4–8½	4–8½
Number of axles (with tender)	14	6	12	12	18
No. of driving axles	8	4	8	8	12
No. of driving motors	8	4	8	8	12
Diam. of driving wheels, in.	44	44	44	44	44
Wheel base of trucks:					
4 wheels (ft-in.)	10–6	—	—	—	-
6 wheels (ft-in.)	14–2	14–2	14–2	14–2	14–2
Distance between truck centers, ft-in	18–11	37–6	37–6	37–6	37–6
Distance between cab centers	57–3	—	—	—	—
Total wheel base of locomotive, ft-in	87–0	51–8	117–8	117–8	183–8
Length of locomotive, alone, ft.	98	66	132	132	198
Length of tender, ft-in	55–6	—	—	—	—
Total length of locomotive with tender, ft-in.	153–6	66–0	132–0	132–0	198–0
Weight of mechanical parts, lb	264,290	172,445	329,890	344,890	502,335
Weight of electrical equipment, lb	137,110	68,555	137,110	137,110	205,665
Weight of thermal plant, lb.	113,600	59,000	118,000	118,000	177,000
Weight of fuel, crew, sand, boiler, feedwater, etc., on locomotive, lb	35,000	30,000	75,000	60,000	105,000
Total weight of locomotive alone, ready for service, lb.	550,000	330,000	660,000	660,000	990,000
Weight of mechanical part of tender, lb	90,000	—	—	—	—
Weight of fuel on tender, lb.	120,000	—	—	—	—
Total weight of tender, lb	210,000	—	—	—	—
Total weight of locomotive and tender, lb	760,000	—	—	—	—
Adhesive weight of locomotive, lb	440,000	220,000	440,000	440,000	660,000
Weight per driver, lb	55,000	55,000	55,000	55,000	55,000
Capacity of fuel on locomotive, gal	500	2,300	6,800	4,600	10,000
Capacity of fuel on tender, gal.	16,000	—	—	—	—
Fuel consumption at full load, gph	534	267	534	534	800
Fuel consumption at one-half load, gph	272	136	272	272	408
Fuel consumption at one-quarter load, gph	174 or 136	87	174	174	261
Thermal-efficiency curve	Figs. 4 & 5	Fig 4	Figs. 4 & 5	Figs. 4 & 5	Fig. 4
Capacity of steam boiler, lb per hr	3,600	3,600	3,600	7,200	7,200
Capacity of water, gal.	3,500	1,100	2,000	2,200	2,200
Traction motor type	GLM735S	GLM735S	GLM735S	GLM735S	GLM735S



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

1—Gas Turbine
2—Turbocompressor
3—Air Preheater
4—Combustion Chamber
5—Reduction-gear Set
6—Main Generator Group
7—Auxiliary Generator
8—Brake-excitation Generator
9—Steam Boiler for Train-heating
10—Storage Battery
11—Auxiliary Diesel Generating Set
12—Electrical Control Apparatus
13—Control Apparatus in Engineer's Cab
14—Air-brake Compressor Set
15—Water Tank for Steam Boiler
16—Fuel-oil Tanks, Main Supply from Tender
17—Lubricating-oil Radiators

Fig. 1. 5000-hp Freight and Passenger Gas Turbine Locomotive with Fuel Oil Tender.

efficiency of 20 per cent without entering the yet largely untried field of gas temperatures above 1112° F. Also, it has been realized that American railroad practice requires heavier electrical equipment, both in generators and traction motors, than is the case in Europe.

The standard power plant of 2500 hp can be used singly or in multiple in a wide variety of combinations. Two specific cases will be considered more closely, as follows:

1. Passenger and freight locomotive of 5000 hp, one unit with two standard power plants for double-end operation: The layout of this locomotive is shown in Figure 1 and its main characteristics are given in Table 1. Particularly worthy of note is that the tender, shown in Figure 1, is not required when the locomotive is in service on short runs. However, for runs of any length, particularly from the Hudson River to the Mississippi River, or from there on to the West Coast, a tender is preferred in order to eliminate stopping for fuel. The size of the tender depends entirely upon the service in which the locomotive operates.

2. Passenger and freight locomotive of 7500 hp in three units, each with one standard power plant operated in multiple: The layout is shown in Figure 2, and the main characteristics are given in Table 1.

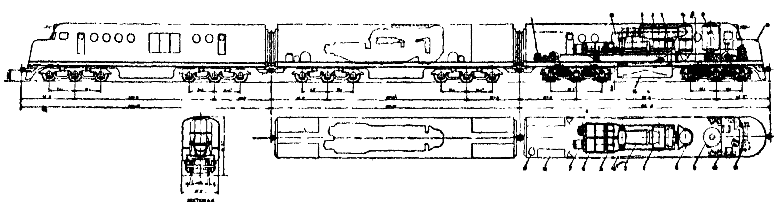
It should be noted that the two end units in Figure 2 (see Fig. 3) are fully equipped for independent operation. One alone represents a 2500-hp locomotive. One end unit together with a center unit forms an articulated 5000-hp locomotive for single-end operation. The two end units coupled together produce an articulated 5000-hp locomotive for double-end operation if this should be found preferable to the design shown in Figure 1.

Thermal Plant of Gas Turbine Locomotives

The prime-mover plant consists essentially of the gas turbine; an air compressor directly coupled to it; an air preheater; a combustion chamber; and the necessary auxiliary equipment, such as fuel-oil pumps, lubricating-oil pumps, speed and fuel-oil regulators, gas turbine shaft-turning de-

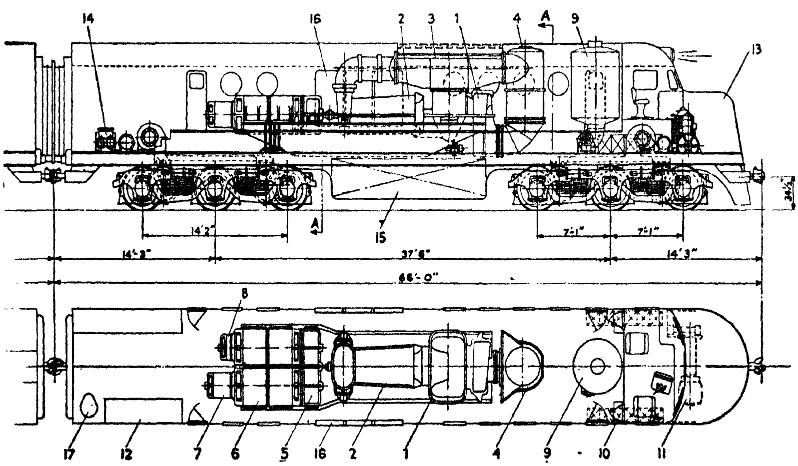
vice, and the necessary safety appliances that are essential for the protection of the unit against overspeed, too high temperature, or lack of lubricating or control oil pressure.

Like any internal-combustion engine, the gas turbine must be started by means of some external power source. In order to accomplish this, a 200-hp Diesel generator set is built into each locomotive. This Diesel generator set itself is started from



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 2. 7500-hp Passenger and Freight Gas Turbine Locomotive.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

1—Gas Turbine
2—Turbocompressor
3—Air Preheater
4—Combustion Chamber
5—Reduction-gear Set
6—Main Generator Group
7—Auxiliary Generator
8—Brake-excitation Generator
9—Steam Boiler for Train Heating
10—Storage Battery
11—Auxiliary Diesel Generating Set
12—Electric Control Apparatus
13—Train Control, etc.
14—Air-brake Compressor Set
15—Water and Fuel Tank
16—Lubricating-oil Radiators
17—Toilet

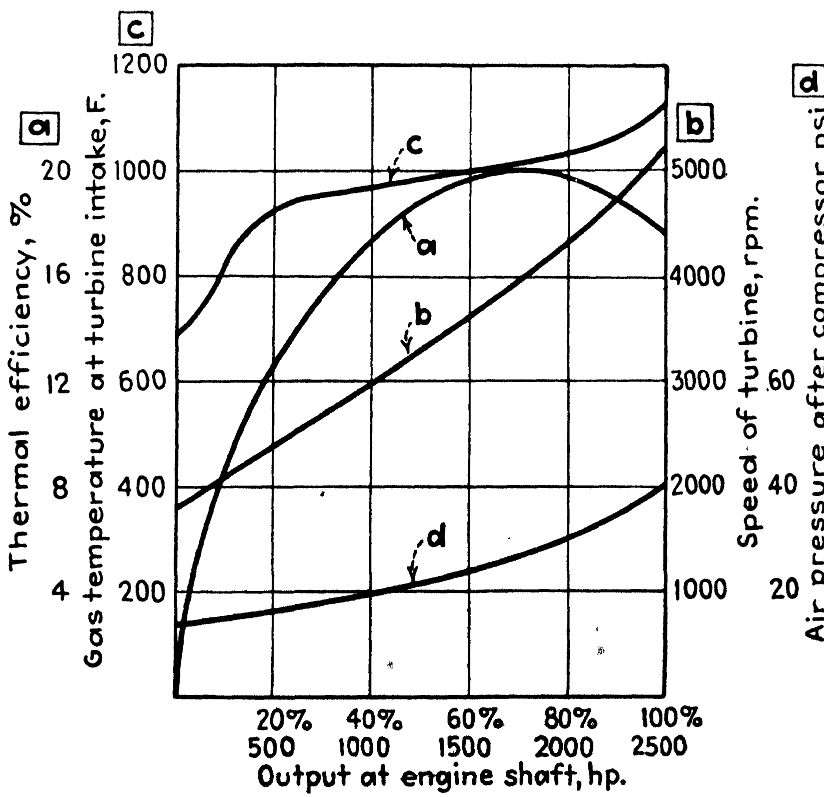
Fig. 3. 2500-hp End Unit of 7500-hp Locomotive of Fig. 2.

the locomotive storage battery, which furnishes power to the generator. The generator then acts as a starting motor.

The Diesel generator group starts the gas turbine group and accelerates it up to a speed where sufficient air is furnished by the air compressor for the combustion of the fuel oil. When this speed is reached, fuel oil is injected and electrically ignited. The turbine now accelerates very quickly to its no-load speed, and the auxiliary Diesel generator can be stopped. The turbine is now ready to operate the locomotive.

The starting time for the gas turbine is approximately five minutes.

The operating characteristics of the gas turbine are shown



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 4. Operating Characteristics of 2500-hp Gas Turbine for Locomotives.

in Figure 4, where the four curves give the following values for the 2500-hp unit:

1. Thermal efficiency in per cent.
2. Speed of gas turbine.
3. Temperature of gases in degrees at turbine intake.
4. Pressure of air after the compressor in pounds per square inch.

Lubricating-oil coolers are provided in the side walls of the locomotive.

Contrary to the lubricating requirements of a Diesel engine, which may reach a high cost, the lubrication cost of the gas turbine compressor unit is practically negligible. Only the renewal of each charge of oil after approximately 10,000 hr of operation is required.

The gas turbine and the compressor unit each contains only two bearings requiring lubrication. There are no crankshafts, no connecting rods, no pistons that move up and down, no cylinder walls to lubricate, no camshafts with their bearings, and no valves. In short, the gas turbine is the simplest of all thermal motive-power units in use today.

This gas turbine is normally operated with Bunker "C" fuel oil, which does not lend itself to ignition at normal temperature. Therefore, a fuel-oil preheater, exposed to the exhaust gases of the turbine, is arranged above the air preheater. The gas turbine is started, and for a few minutes is operated with Diesel oil. When the Bunker oil has reached a temperature of about 175 to 200° F, the change-over to Bunker oil is made.

The output of the gas turbine is regulated by setting its speed at certain values. The master controller, which influences the governor, accomplishes this result. The speed governor, in turn, controls the fuel supply to the combustion chamber in such a manner that the speed for which the master controller is set is obtained at the turbine shaft. The output-control system of the locomotive is simple and functions in such a way that, over a wide speed range of the locomotive, the output at the turbine set is practically constant for each selected speed

point. A temperature-control device is arranged to decrease the load if the normal operating temperature of the turbine is exceeded, and to increase it if the turbine operates below normal temperature. With this arrangement, the gas turbine always works with its best efficiency.

Unlike a Diesel engine, sudden heavy overloads of the gas turbine do not stop it: they merely lower the speed of the machine until the output-regulating device has had time to adjust the fuel-oil supply and the excitation of the main generator.

The no-load speed of the gas turbine set is about 1820 rpm, whereas at full load, it operates at about 5200 rpm.

Reduction-gear Set

Since it is not desirable to operate the generators at 5200 rpm, a reduction-gear set is arranged between the gas turbine compressor shaft and the generator group.

Electrical Equipment

Investigation of methods of transmitting power from the prime mover to the wheels of the locomotive has led to the adoption of electrical transmission as most suitable, for the power range involved, for easy and smooth regulation at all speeds. In the 2500-hp units, the direct-current generator group furnishes power to four forced-ventilated series-type traction motors.

Overload relays and motor switches protect the motors and the generator from dangerous overloads. Field weakening of the traction motors is employed in order to cover the full-speed range of the locomotive without reaching unduly high voltages on the generator.

The generator group consists of four armatures, each delivering power to one traction motor. No series-parallel connection is used on the 2500-hp unit, the motors being always connected to the same generator armatures over the entire speed range of the locomotive. The absence of such a series-parallel arrangement simplifies the locomotive and assures smooth power control over the entire speed range.

The generators are built as differential machines having both a self-excited and a separately excited winding. A starting winding is used to drive the gas turbine set from the auxiliary Diesel generator set when starting. Output regulation is accomplished mainly by regulating the separately excited winding, which, in combination with the differential winding and the temperature-control device of the turbine, maintains a constant power output of the group for various locomotive speeds. For the operation of all auxiliary circuits, such as battery-charging, traction-motor ventilating groups, fuel and lubricating-oil pumps, and so forth, an auxiliary generator of 65-kw rating is directly coupled to the main generator. A rocking-type voltage regulator is used to keep the voltage of the auxiliary generator constant for all speeds from no load to full load.

Electric Braking

The gas turbine locomotive lends itself readily to electric braking; in fact it is ideally suited to it. During braking, the motors furnish power to the generator (instead of to resistors, as would be necessary for other types of locomotives), which in turn drives the gas turbine set. The air compressor absorbs all power that is delivered by the generator. The locomotive will be able to brake any train on a downgrade that it can haul up the same grade.

Traction Motors

The traction motors, which are of the forced-ventilated type, can be either of the nose-suspended or entirely suspended, rigidly mounted type. In the latter case, none of their weight rests on the driving axle. A flexible disk drive is then used to transmit the motor torque to the driving wheels.

These motors have already been used in single-phase locomotives with pulsating motor torque and power of up to 1000 hp per axle. They permit the free, vertical movement of the axle within the limits of the journal-box guides. Most important of all, this disk drive does not require any axle lubrication other than the usual journal bearings, and does

not influence the springing of the locomotive. The application of this very simple unit is strongly recommended because it protects the motor from the heavy blows otherwise imposed upon it from the axle, and it reduces the unsprung dead weight to a minimum.

As has been previously mentioned, the locomotives are equipped with an auxiliary Diesel-generator set of approximately 200-hp rating in order to start the gas turbine. This set is also used to feed one of the traction motors when it is desired to move the locomotive only a short distance at slow speed, such as moving it out of a shed, or to a train for coupling up, before the gas turbine has been started, or even for light switching work.

Stopping the Gas Turbine

The gas turbine is shut down by stopping the fuel-oil pump. In order to prevent the bending of the turbine shaft during the cooling-down period, a shaft-turning device is installed in the locomotive. A timing relay controls a shaft-turning motor of small output (fed from the storage battery) in such a way that the turbine shaft is turned ½ revolution about every 30 min. This continues for a period of about six hours.

Mechanical Parts

The mechanical parts proposed for these engines are in accordance with standard American locomotive-construction practice.

The cab rests on a heavy underframe which is either of welded or integral-cast construction. The 2500-hp units rest on two trucks which are the six-wheel type for freight and high-speed passenger locomotives. None of the trucks is articulated—that is, the tractive pull of the locomotives is transmitted through the cab underframe.

As shown in Figure 1, the 5000-hp locomotive rests on four trucks, of which two have four, and the other two, six wheels. The six-wheel trucks and the motors installed in them are the same as those used in the 2500-hp units. Only the springs will be different on account of the different axle load. Otherwise,

these trucks are interchangeable for the proposed locomotive types.

The trucks are of the four- and six-wheel spring-bolster type having integral steel-casting frames. The axle boxes can be either of the antifriction or sleeve-bearing type. Two traction motors are mounted in each truck, the middle axle of the six-wheel truck being an idler. Since the motors are of the forced-ventilated kind, a flexible air connection is arranged between the truck and the cab underframe, on which the motor blowers are located.

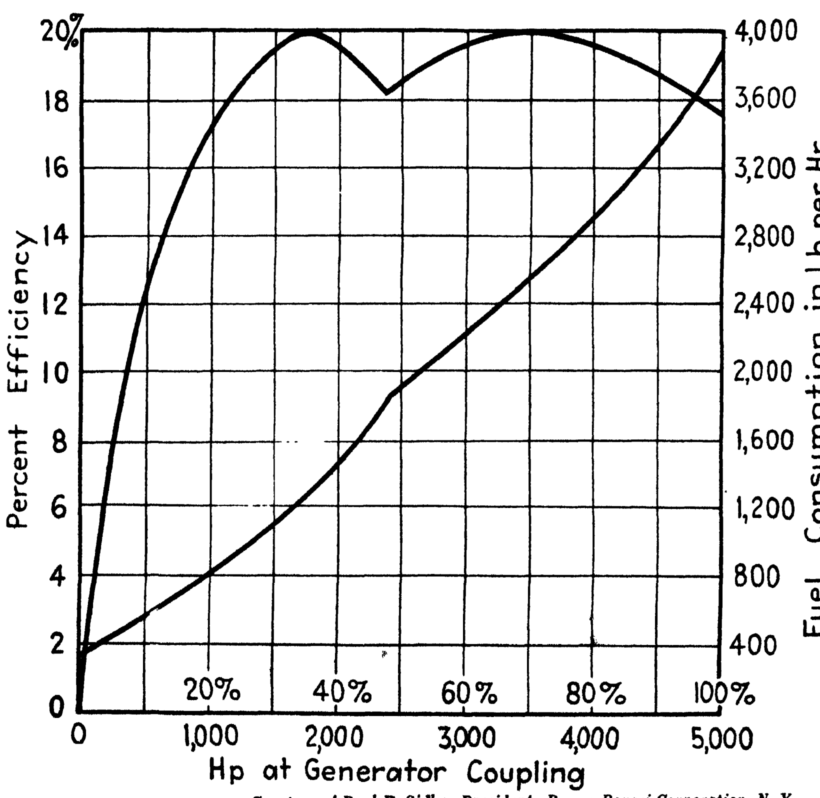
All these proposals contemplate electrical transmission. Aside from its long experience in this field, Brown Boveri also believes that in introducing a novel type of locomotive power plant, it is important to eliminate any causes for disturbance which might obscure the main issue. Mechanical transmissions for the power and speed conditions necessary on a locomotive do not appear to have been used successfully for sufficiently long periods to warrant their immediate consideration. This may change at a later date after more developmental work has been completed. The electrical transmission also has the important advantage of allowing a much greater flexibility in weight distribution, particularly in limiting the axle load.

Thermal Efficiency

It is a well-known characteristic of the gas turbine that, in order to obtain high thermal efficiency at fractional loads, the speed of the compressor must be varied according to the load. When the compressor is coupled to the gas turbine on the one side and on the other side to an alternating-current generator with constant frequency, or a mechanical device requiring constant speed on the driving shaft, this desirable speed variation cannot be obtained. For such cases, Brown Boveri has built two-shaft arrangements comprising the one gas turbine driving the air compressor only and allowing variable speed, and a second gas turbine, rated for the net power, which drives the generator or the mechanical device at constant speed. For a locomotive power plant with direct-current electrical transmission, this two-shaft arrangement is not nec-

essary, since it is readily possible to vary the speed of the single-shaft set in accordance with the load.

Figure 5 shows the thermal efficiency of both power units of the 5000-hp locomotive, as shown in Figure 1, when one unit carries all the load up to 2400 hp. In order to obtain this high efficiency at low loads, it is necessary to have each of the four generator armatures of the first unit supply power to two traction motors: when the second unit comes in to supply power above 2400 hp, each of the four generator armatures will furnish current to only one traction motor. However, if the two-shaft single-unit 5000-hp turbine was used in Figure 1, the electric controls would be as simple as the present 2500-hp unit, each generator armature feeding two motors at all times.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 5. Efficiency and Fuel Consumption for 5000-hp Locomotive of Fig. 1.

On the other hand the two-shaft single-power-plant arrangement requires further mechanical controls for the gas turbines in addition to those of two separate complete power units, operated in parallel or singly as the service may require.

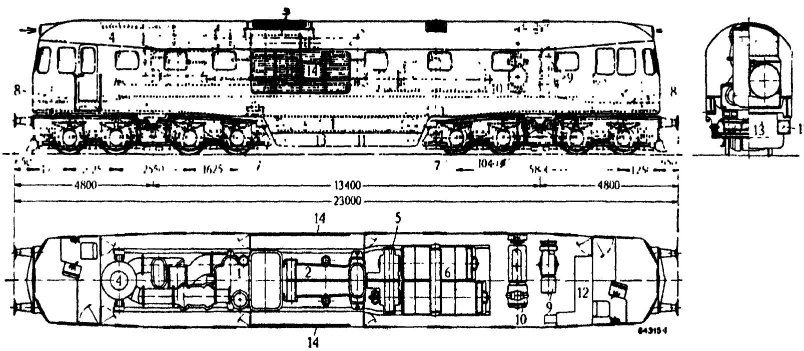
4000-hp Unit

The locomotive of Figure 6 has the latest Brown Boveri gas turbine power plant. This unit delivers 4000 hp to the two main generators. The generators are driven by a reduction gear set in the same manner as shown in Figure 3. The gas turbine unit has a main power shaft very similar to that in Figure 3, and in addition, it has an auxiliary booster which raises the air pressure up between the main compressor and the combustion chamber. The 4000 hp is based upon 650° C temperature of the gases into the main turbine. Brown Boveri has established the policy of holding the temperature of the gases into the turbine down to 600 or 650° C in order that the turbine might have a long life. If higher temperatures were used in this unit, it would of course produce more horsepower without changing the unit's present dimensions.

Particularly to be noted is that the construction of the locomotive shown in Figure 6 is for European use, and is not suitable for American railroads. One suggestion for meeting American conditions is to place the power plant shown in Figure 6 into the unit shown in Figure 3, and then adding a tender or auxiliary unit as shown in Figure 1. This arrangement would place all the fuel and the train heating boiler and its water into the tender, and the size of the locomotive would be determined by the nature of the service into which it would be placed. In this case, the power-plant unit and the tender would each have four traction motors.

The 4000-hp power-plant unit shown in Figure 7 could be placed in the cab shown in Figure 1, and there would be room for a limited supply of water and fuel without using the tender. This unit has an expensive construction, there is no saving in weight, and two four-wheel and two six-wheel trucks are still required.

In conclusion, it should be noted that further analysis is

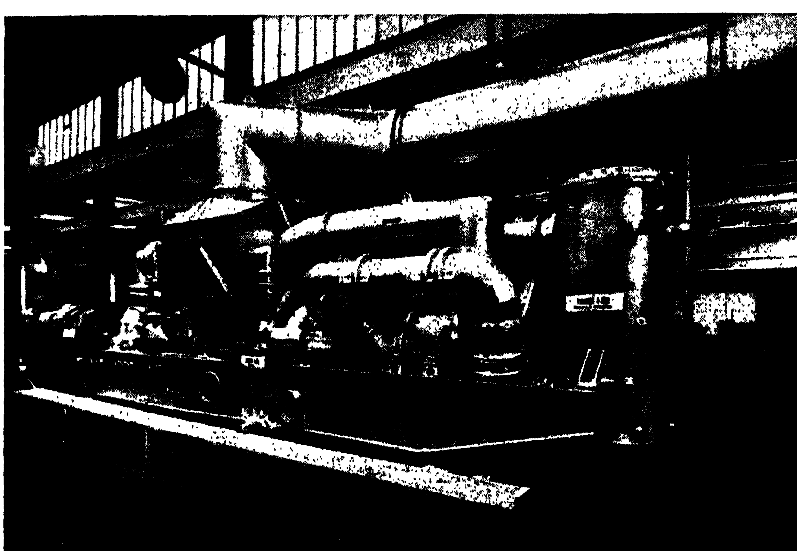


Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

1—Turbine
2—Compressor
3—Air Preheater
4—Combustor
5—Reduction-gear Set
6—Generators
7—Traction Motors
8—Traction Motor Blower
9—Air Brake Compressor
10—Auxiliary Diesel Generating Set
11—Storage Battery
12—Control Apparatus
13—Fuel and Lube Oil Tanks
14—Lube-oil Radiators

(Comprex shown between #1 and #4)

Fig. 6. Brown Boveri 4000-hp Gas Turbine Locomotive.



Courtesy of Paul R. Sidler, President, Brown Boveri Corporation, N. Y.

Fig. 7. Brown Boveri 4000-hp Gas Turbine Power Plant. Reading from left to right are two generators, compressor, regenerator, turbine, booster unit and combustor.

required to perfect a gas turbine locomotive design for American conditions. There are many incidental problems to be solved even if the locomotive burns oil, and more problems if the locomotive burns coal. In any case, Brown Boveri believes that the electrical equipment, the mechanical construction, and the gas turbine power plant—each taken separately—can be built to give reliable performance. However, the combination requires considerable study, not so much on paper as on the road, in actual service in America, as has already been done in Switzerland.

Chapter VI

Elliott Gas Turbine Unit[1]

Layout of Experimental Gas Turbine Unit

An Elliott plant composed of two turbines, two compressors with intercooling, two combustion chambers, and a regenerator is shown in Figure 1.

The flow of gas in this unit begins at the low-pressure compressor which takes in free air and compresses it to a pressure of 43 psia and 300° F. The temperature is then lowered in the intercooler, whereupon the air passes directly into the high-pressure compressor which raises the pressure to 96 psia. The air then passes through the regenerator, where a portion of the heat in the exhaust gas is recovered before it enters the high-pressure combustion chamber.

In the high-pressure combustion chamber, fuel oil is burned directly in the air stream, and a temperature of 1230° F is reached at the entrance to the high-pressure turbine. In this turbine, the heated air is expanded to 53 psia, and thus sufficient power is developed to drive the low-pressure compressor.

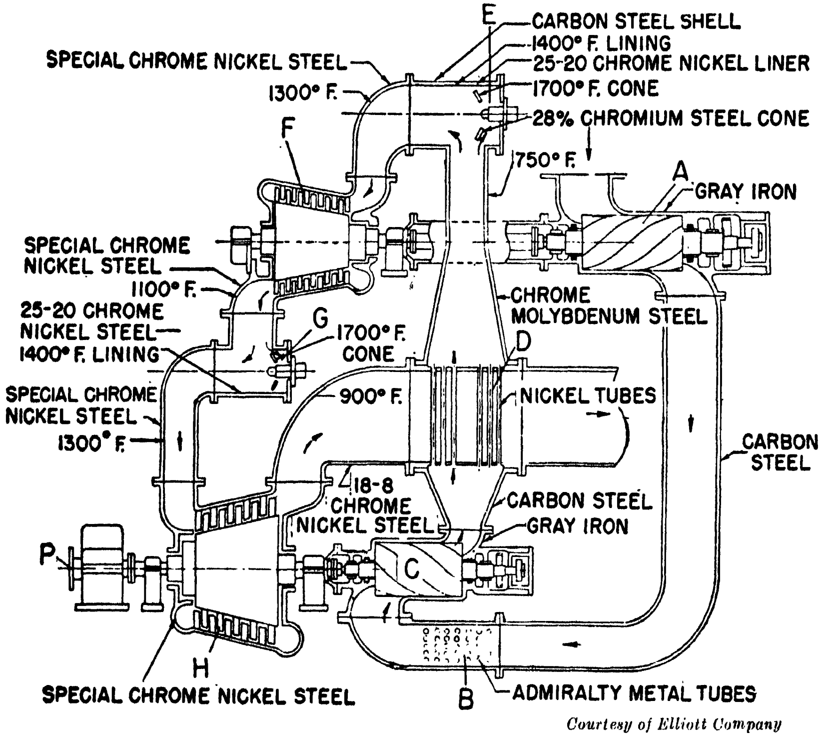
The air from the high-pressure turbine exhaust is then reheated in the low-pressure combustion chamber to elevate its temperature to 1207° F before it is expanded in the low-pressure turbine. Five thousand horsepower is realized from the low-pressure turbine, 2500 hp of which is expanded in driving the high-pressure compressor. The remainder is excess power which, in the case of a marine gas turbine, drives the propeller.

After the air leaves the low-pressure turbine at slightly above atmospheric pressure, it passes to the regenerator

[1] Abstract from *Powerfax* magazine of the Elliott Company.

where it preheats the fresh, compressed air from the high-pressure compressor. The exhaust of 400° F is discharged to atmosphere.

The general arrangement, as shown in Figure 1, indicates that the two compressors are driven by separate turbines. This is not an essential feature of a plant containing two turbines and two compressors, but it is a desirable one which in this



Courtesy of Elliott Company

Fig. 1. Diagrammatic Layout of Gas Turbine Plant.

case was selected for a particular reason. Naval power plants operate more than 90 per cent of the time at other than full load, and it is therefore important to have highly efficient part-load performance. The arrangement of machines in a gas turbine cycle, with individual turbine drives for the compressors, makes it possible to achieve this result.

To understand why this is true, it is necessary to know why part-load and full-load economy should be different. To

reduce the power output, it is necessary either to decrease the amount of hot air passing through the turbines or to reduce the temperature of the air; but it may be necessary to do both.

Best efficiencies are realized when the temperature of the compressed air is as high as possible before the air is put to work in the turbines. Thus, for best economy, it is desirable to operate at reduced power by decreasing the supply of compressed air rather than by reducing the gas temperature. The best approach to high-temperature, reduced-flow operation is realized in this type of plant because the components are so arranged that the main power turbine can always be operated at full temperature.

Complete control is obtained by regulating the fuel flow to the turbine driving the first-stage compressor. Since the amount of air that enters the system is controlled by this compressor, it is apparent that this one feature can produce ease of control and, at the same time, permit efficient operation of the main power turbine.

A major problem of elastic design must be solved in setting up any group of machines, some of which are operating at 1200, and others at 100 and 300° F. It is axiomatic that metal expands when it gets hot, and that the hotter it gets the more it expands. This situation is particularly acute with a gas turbine installation since, in addition to the fact that the temperatures are very high, the turbines must be made from materials which have a coefficient of expansion 50 per cent higher than ordinary steel. For example, the turbines are nearly ½ in. longer at operating temperature than when they are cold, and this requires designing these machines, their mountings, and all parts connected to them in such a way as to permit expansions caused by heat to take place in a short space, without pushing the machines from their foundations or breaking the connecting pipes. No thermal strains should be imposed on the various machines, since the lightweight structures which are subjected to high temperatures may be seriously distorted.

Both the high- and low-pressure turbines are solidly

mounted at their exhaust ends, with all expansions taking place toward the inlet of the machines. In order to permit this expansion, these floating ends are secured by means of freely moving links. Relative movement between the turbines and the compressors they drive is absorbed by double flexible couplings and the long, torque tube jackshafts.

Other temperature-expansion problems are found in the pipes which connect the various units of the plant. Flexible joints of conventional type are either not satisfactory for the temperatures and pressures encountered, or are too large to be used in this type of plant. Therefore, a new type of expansion joint was required.

The temperature at the inlet of the high-pressure turbine reaches 1230° F, yet at a point about 10 inches away, the metal temperatures must be below 200° F for proper operation of the main shaft bearings. In order to make this possible, special methods were employed to prevent the free conduct of heat to undesired locations, which resulted in the use of systems of heat dams and special oil cooling at the bearings, and air cooling at the pin rings. The pin rings at each end not only support the full weight of the turbine, but are so constructed that a temperature drop of 600° F takes place in the connections between them and the inlet casing, a result that is accomplished by using small areas of contact, reduced metal sections to cut down heat conduction, and some well-placed air cooling on the inside of the pin rings.

A further temperature drop takes place in the bearing casings, and here again a problem exists. The inside flange of the bearing casing reaches 550 to 600° F, but five inches away, the temperature can be only about 200° F, thereby requiring symmetrical and uniform sections for the walls of the casing to prevent excessive and uneven temperature strains.

The shaft must be designed in a similar manner. The stub shaft ends have reasonably light walls so that the heat conduction is kept to a minimum. A radiation shield is installed inside the shaft to prevent exposure of the cool end of the shaft to the red-hot inner regions. In addition to this, oil cooling is provided under the bearing journal sleeves.

One other outstanding feature of this gas turbine design is evident in many locations. All main members are tied together by means of radial pins which permit a free movement between adjacent parts that experience temperature differences. In other words, the pin rings, the diaphragms, and the gland assemblies are all free to expand unrestrainedly with respect to their adjoining members. Such a heat-elastic design and construction has eliminated many of the disastrous problems that confronted former builders of gas turbines.

A method was needed for getting the machines unbolted after they had been operated at temperature. The stainless-steel alloy which must be used in these turbines has the undesirable characteristic of galling when two parts are assembled together. Under such conditions, it is impossible to take apart the nuts and bolts unless a suitable compound has been used previously on the threads. Many such compounds are sold, but none was satisfactory for the temperatures at which the Elliott Company plant operated. Therefore a special colloidal silver compound was devised.

Some Manufacturing Problems

Major problems in designing a successful gas turbine plant are materials for operation at the high temperatures required. An examination of the temperatures encountered throughout the cycle of the Elliott gas turbine plant will, therefore, be instructive.

Two machines—the high- and low-pressure turbines—and a considerable part of the duct work operate at a temperature high enough to make the steel visible in the dark—that is, at a red heat. The physical characteristics of metal at these high temperatures are a problem in themselves.

Not only is the strength of the metal reduced at high temperatures, but in addition, the phenomenon known as *creep* becomes apparent when exposure to high temperature is prolonged. Creep exhibits itself as a growth, or elongation, of the material when it is subjected to load, and the rate at which this elongation takes place is governed by the material, the load applied, and the temperature.

Because of creep, it is certain that, after some period of operation, the turbine rotors will grow, the flat-sided ducts will bulge, and the round ducts will grow too large and too thin. Therefore, the designer must choose materials and loadings of such character that changes caused by creep will not be obnoxious before a certain definite time in terms of hours of operation. The present Elliott Company plant is designed for 10 years of continuous high-temperature service.

Figure 1 illustrates a schematic cycle showing materials and temperatures in all major parts of the plant. A fair gamut of materials has been run on this job. The nickel toroidal joints in the high-pressure combustion-chamber inlet were an interesting problem. This duct is made of chrome-molybdenum steel to operate at temperatures up to 1000° F. The toroidal joints are spinnings, 0.025 in. thick. If made of chrome-molybdenum steel, the spinnings would be subject to scaling, which would be dangerous in that the material is already of minimum thickness. Austenitic stainless steel, having a higher coefficient of expansion than chrome-molybdenum, would cause intolerable differential stresses, and ferritic stainless steel cannot be spun successfully. Copper alloys have high coefficients of expansion and very poor high-temperature properties. Nickel is usually a work-hardening material which cannot be spun, but a very small quantity of a special grade of nickel was produced which could be spun.

The use of high-temperature materials creates so many manufacturing problems that the only possibility of successful construction comes through extremely close cooperation of the manufacturing design departments. Castings, though simple and convenient to design, are hard to produce in high-temperature alloys and do not have the high-temperature properties of rolled or forged material of the same analysis. Because riveted joints depend primarily on tension in the rivet, they can be used only in minor attachments.

In general, the only recourse in building such machines as this is to use rolled plate and arc welding, and by this method to fabricate many pieces into one permanent, single assembly. This method of fabrication was used in all of the duct work

and the combustion chambers in this gas turbine plant. The turbine rotors were assemblies machined from rolled plate and forgings, and welded into an assembly.

The extended use of welding in the construction of a gas turbine plant brings up some interesting problems in connection with welding on materials fit for high-temperature service. As an instance, SAE 4130 chrome-molybdenum steel is an air-hardening variety. Danger exists that when a weld is made, the heat of the weld will create an extremely hard, brittle zone directly adjacent to the weld. Brittleness is more pronounced in heavy than in light material owing to the quicker cooling of the welds. As a result, it is necessary to consider each weld, to set up procedures, and to determine whether or not preheating is required.

The welding of 19–9 WMo material was a completely new problem. No welding had been done on the alloy prior to the design of this gas turbine plant. A new welding electrode was required because all existing electrodes were tested and found to be short of the necessary high-temperature strength.

Furthermore, since there were some very difficult welds on the job, tests were necessary to ascertain the operating characteristics of the electrode. When the best operating characteristics had been learned, extended tests were made to discover the best procedures to use in laying weld metal. During these tests, trouble encountered in connection with cracks in the welds was run down and found to be a matter of analysis of the core wire used in producing the electrode. Without such preliminary experimental work, it is doubtful whether the rotors for the high- and low-pressure turbines in this plant would ever have been built.

In designing the rotors for the large Lysholm compressors used in this plant, it was necessary to use steel shafts. However, steel was not a good material for the rotors themselves, and the problem was posed of how to attach the steel stub shafts in the cast-iron rotors. A number of methods were studied, but the most advantageous from the point of design appeared to be by a low-temperature braze. Low-temperature brazing, or silver soldering, was a well-known process that was

used extensively on small parts, but seldom employed on parts as large as these rotors. Because the process appeared attractive, tests were made to learn under what circumstances predictable joints could be made in large parts. As a result of these tests, the brazing material was properly placed, a method of cleaning and fluxing was developed, and a heating cycle procedure set up which assured good joints.

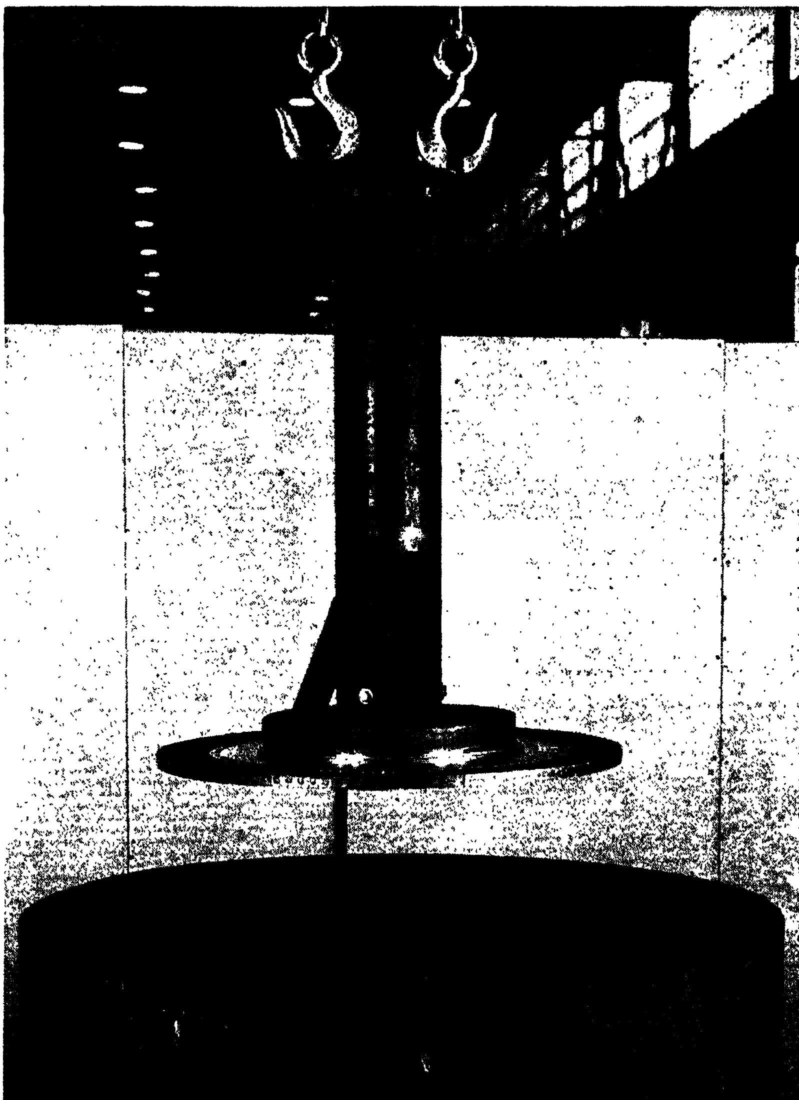
In the manufacture of the regenerator, it was necessary to have a large number of joints which would transmit heat from the tubes to the fins. Some conception of the number of joints required can be gleaned from the fact that in this regenerator there were over 8½ miles of nickel tubing, the largest order of nickel tubing ever supplied. These joints had to remain strong at a temperature exceeding 1000° F. With considerable testing and numerous changes, a copper brazing process was developed for building the regenerators from nickel tubing and sheets. The copper-brazed joint as used is a good carrier of heat; is relatively inexpensive; and when the brazing job has been completed, the parts are clean and free of scale.

Probably the most unusual welding job in the whole gas turbine plant was the fabrication of the turbine rotors. As has been indicated above, the rotors were fabricated from parts machined from forgings and rolled plate. The rotor disks were completely machined and the blades were attached and shrouded prior to assembly of the rotor.

The stub shaft for the inlet end of the rotor was set up in a framework, and the first disk was heated in hot water and placed on the stub shaft. Pressure was exerted to hold it tightly against the stub shaft during cooling. When cool, this disk was shrunk securely to the stub shaft. By the same process, each of the succeeding disks was shrunk onto the rotor assembly.

Figure 2 shows the rotor welding frame in various stages of the rotor assembly. When the rotor had been completely assembled in the vertical position, four tack welds were made in each welding groove to hold the parts together and to apply tension to them in an axial direction. At this point, rollers

were applied to hold the rotor, and the whole frame was laid over on its side, putting the rotor in a horizontal position. A pulley was then attached to one end of the rotor and belt-connected to a variable-speed drive, which would rotate the

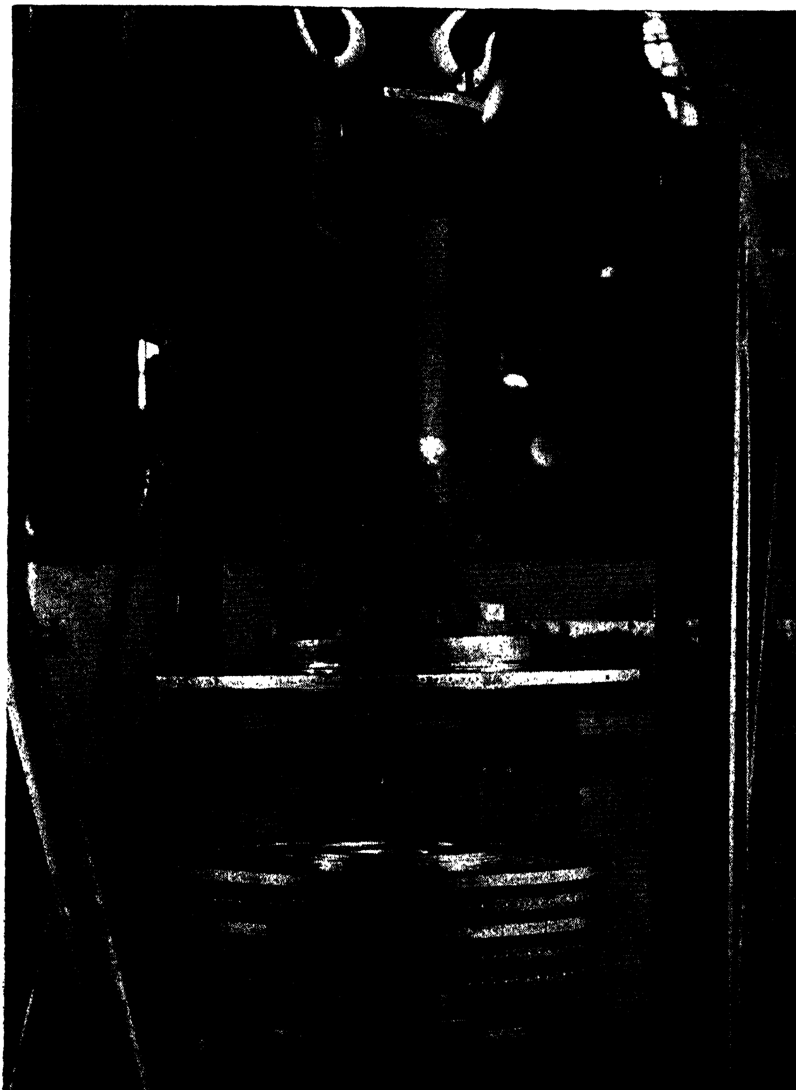


Courtesy of Elliott Company

Fig. 2. Stage in Assembly of Turbine Rotor in Welding Frame.

rotor at welding speed. Figure 3 shows the rotor in the frame ready for welding.

Guards were put between the various disks to protect the blading and disks from weld spatter, and welding was carried



Courtesy of Elliott Company

Fig. 2. Stage in Assembly of Turbine Rotor in Welding Frame.

out to a very set sequence laid out on a calendar and checked off as each weld was made. When welding was finished, the rotor was removed from the frame and placed in a furnace in which it could be rotated at a slow speed and heated to

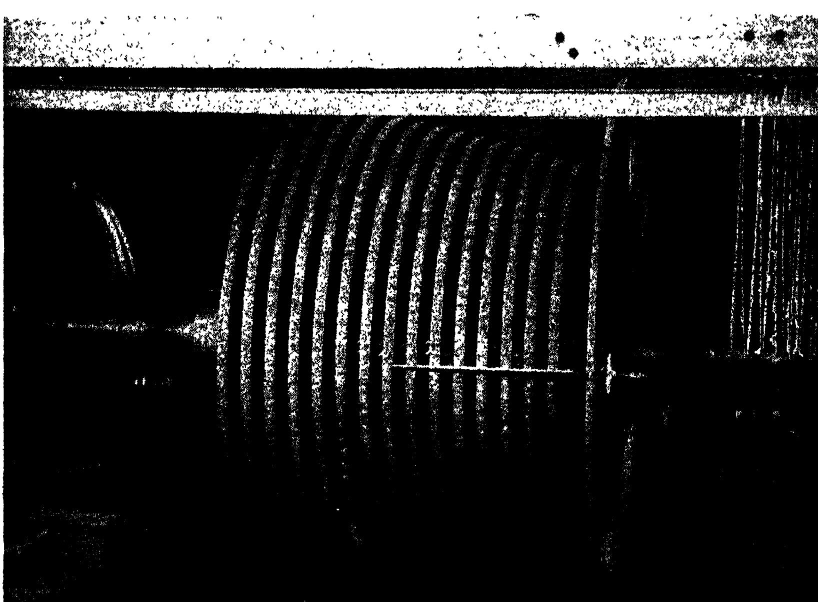
Courtesy of Elliott Company

Fig. 2. Stage in Assembly of Turbine Rotor in Welding Frame.

1400° F. This temperature was maintained for 16 hr, and then the rotor was allowed to cool. Some rough machining was done on the rotor before it was returned to the furnace to undergo a process known as *heat indication.* All heat-treating on this rotor was done in the special furnace (Fig. 4).

In some cases, it was found that rotors operating at high temperatures will bend, and straighten out when cooled. This means that a rotor, in perfect balance when cold, will not be in balance at high temperature. To eliminate this defect, the rotor is heat indicated, or placed in a furnace with the temperature increasing gradually while it is being rotated. As the temperature is increased, the amount that the rotor bends is gauged by measuring the eccentricity with dial indicators.

When the rotor reaches a point at which there is no increase in bending with increase in temperature, it is considered to have been heat indicated and to be set in a stable condition; thereafter, it will not bend when the temperature is

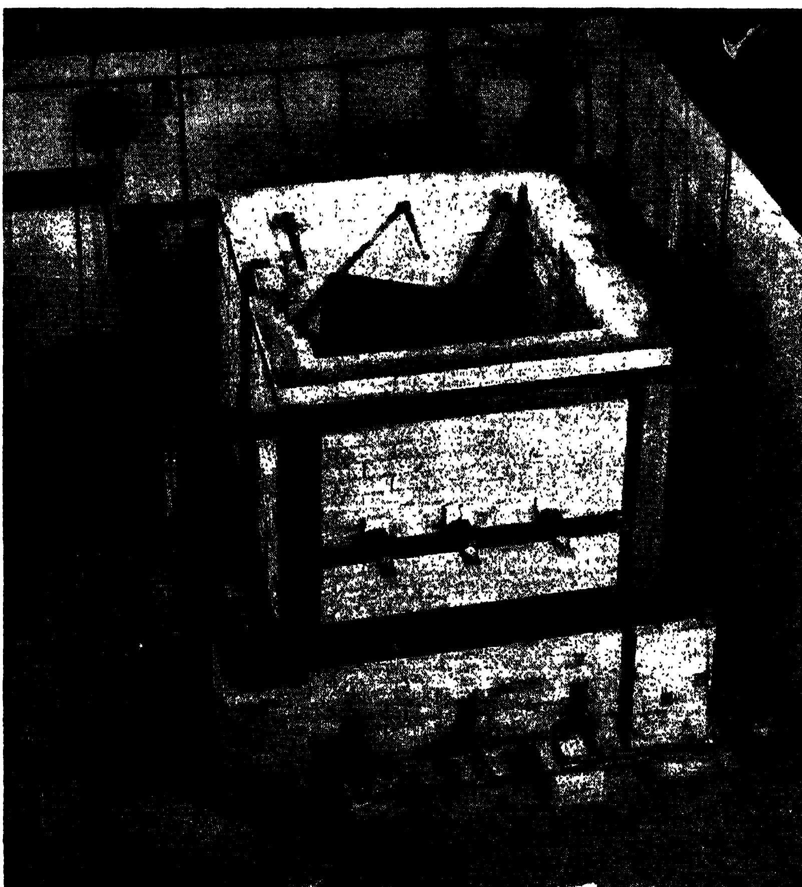


Courtesy of Elliott Company

Fig. 3. Turbine Rotor in Frame, Ready for Welding.

changed. After heat indication, the stub shafts of the rotor are finish-machined, and the rotor is ready for installation in the turbine.

Only a few of the manufacturing problems encountered in building equipment for operation at exceedingly high temperatures have been described. As the temperature at which gas turbines are to be operated increase, these problems will become more difficult. However, it is felt that the biggest gap to be bridged has been the building of the first machine, and



Courtesy of Elliott Company

Fig. 4. Rotor in Special Furnace.

that the experience gained can be carried over, with slight modifications, to the building of future gas turbine power plants.

Starting and Operating a Gas Turbine

In all power plants (Figure 5 is shown as an example), there is a certain amount of routine checkoff and inspection to be performed prior to starting. However, far less preparatory work is needed to get this gas turbine under way than is required for a comparable steam plant. The sole requirements are to engage the starting motor, light off *one* burner, and warm up. The present dollar-wise policy followed in the warming-up period arises from insufficient information on just how fast the gas turbine can warm up. First consideration is being given to obtaining all data necessary for the advancement of gas turbine design. With this out of the way, second-

Courtesy of Elliott Company

Fig. 5. 2500-hp Marine-propulsion Test Unit.

ary problems will receive greater attention than is now being given them. However, it is known that only a slightly longer period is needed to warm up a gas turbine than would be the case if steam were being used.

An examination of the control-panel layout will show that the plant operator has all the operating equipment within easy reach. He has complete control of the entire plant. In an instant, he can change the power developed simply by adjusting the combustion rate. The operator can speed up or slow down, go into regions of very light loads or stop the plant entirely, without moving from the control panel. He needs no assistance from either of the other two men comprising the operating personnel.

From the time fuel is first admitted to the combustion chamber, the warming-up period is merely one of inspection of the plant by crewmen. The operator, having all operating conditions indicated on the control panel, governs the rate of warm-up by means of the throttle lever.

When the inlet temperature on the high-pressure turbine has been slowly brought up to the point at which the starting motor is to be dropped, the temperature on the low-pressure turbine will also have risen slowly to an acceptable value. The two by-pass valves are operated electrically by push-button control on the *main* panel, and at this time, these two remote units are energized in proper sequence to cause them to close, and thus synchronize the plant. As these valves close, the speed of the low-pressure turbine high-pressure compressor shaft increases from the 250 rpm maintained throughout the warming-up period to approximately 900 rpm. The operator raises the starting-motor speed and increases the fuel-combustion rate to compensate for increased air flow. At just about this time, the plant *floats* off the starting motor and is now in a condition to develop useful power over and above that needed to run the compressors, which is often referred to as *back work.*

The remaining burner in the high-pressure combustion chamber is now *lit off,* and there is an increase in power developed. When the burner in the low-pressure chamber is lit

off, there is likewise an increase in power output. With both burners in the high-pressure chamber going at full rate and the low-pressure chamber secured entirely, an output of 2000 hp can be obtained. Furthermore, there is reason to believe that the unit can be run on very light loads by securing both burners in the high-pressure chamber and using only the one burner in the low-pressure chamber. That would mean running the high-pressure turbine on the preheat available, and is yet to be investigated thoroughly.

In a steam plant, to go into a full-power condition immediately after the warming-up period is most unwise, but such a procedure can be followed in this plant. When the warming-up period is completed, operating temperatures are within 100° F or so of the temperature set by the materials used in building the turbines. Thus, maximum power output is a matter of a few moments on securing the starting motor.

From the time the starting motor has been secured, the power output realized in all subsequent settings of the controls is a function of combustion rate and low-pressure turbine speed.

Having already considered the construction of the two combustion chambers and their operation, it will be seen that they are one of the main reasons why one man can completely control this plant throughout its entire operating range without leaving the throttle board. Unlike handling a steam installation, with its attendant boilers, the operator is never compelled to change burners or burner tips in order to maintain a combustion rate commensurate with power developed. This is a radical improvement in itself.

Change in load level does not materially alter the inlet temperatures to the turbines—only by the variation of combustion rate do we increase or decrease the power developed. When, in operating at any level within the power range of this plant, an increase of power is necessary, the operator can set his throttles in a matter of seconds. If he is increasing power, he advances his throttles on all burners until inlet temperatures begin to climb above upper limits, and almost instantly the effect is felt throughout the entire unit.

What happens is that the operator gives a slight impetus to the system by the excessive advance of the throttles. The *excess* condition is held only momentarily, its effects being canceled out after a brief period of operation.

The increase in temperature has caused an increase in speed; this causes an increase in air flow, which in turn tends to reduce the temperature that brought about the original change of conditions. Except for this manner of changing load, the turbine unit operates at constant temperature, since the momentary temperature variation is approximately 8.3 per cent of the designed operating value. This small variation of temperature exists for so brief a period that the actual heat level within the mass of the turbines is relatively unchanged, and thus does not accelerate metal fatigue.

Stopping the unit is accomplished by closing all throttle levers to the *off* position, stopping the injection motor, opening the by-pass valves, and engaging the jacking gears when the speed on both shafts drops to approximately 10 rpm.

Chapter VII

The Closed-Cycle Gas Turbine Unit[1]

Typical Components of Closed-circuit Plants

Although the duty of the various machines and apparatuses remains fundamentally the same in all installations, various designs nevertheless result which are dependent on the different outputs required, the uses to which the installations shall be put, the efficiencies, and the available fuels. The closed cycle operating at a pressure above atmospheric leads to conditions of construction which deviate considerably from other kinds of gas and steam turbines proposed by Ackeret & Keller in 1935. The consistent application of knowledge gathered from the laws of flow, from up-to-date aerodynamics, and concerning the improved properties of metals subjected to high temperatures, coupled with proper harmonizing of the various components, has in recent years led to ever-increasing simplicity for designs.

The designs deviate in many respects from the layouts usually encountered for turbomachines. A closed-circuit plant is not simply an assembly of known components, but is the arrangement of the various parts in relation to one another. The course of the working medium through the whole installation has been the subject of careful study, and due attention has been paid to heat expansion. Only in this way can the pressure losses in the piping and the other secondary losses be reduced to an admissible measure.

In this connection, it has proved very advantageous that all parts of this plant, such as turbines, compressors, heat ex-

[1] Revision of report presented to A.S.M.E. by Dr. C. Keller, Director of Research and Development, Escher Wyss Works, Zurich, Switzerland.

changers, and air heaters form part of the actual manufacturing program of Escher Wyss, who specialize in the construction of turbomachines of all kinds. Thus, all the components have been developed and built in the company's own works. The experience acquired in various fields was coordinated, and proper use made of the latest research results obtained in the company's own hydraulic and caloric laboratories.

These installations are suitable for standardization. Thus, the whole range of stationary installations from 3000 to 50,000 kw can be dealt with by a few types of machines and apparatuses. The component parts of the heat exchangers themselves as well as the regulating means can, for installations of these various sizes, be put together by suitable combination. Such standardization which has been attempted, but never realized, in the case of steam turbines will have a favorable influence on price calculations.

Turbines and Compressors

Since the specific volume is reduced owing to the raised working pressure of the closed circuit, the machines are astonishingly small when compared to open-circuit combustion turbines. In addition, the heat drop that has to be dealt with is much less than in the case of steam turbine plants, so that the closed-circuit turbine has not many stages.

For informatory purposes, the main dimensions of the rotors for standard closed-cycle plants of various outputs have been indicated in Table 1. These dimensions may vary slightly according to the speed and blading (action or reaction) of the

TABLE 1

APPROXIMATE FIGURES FOR MACHINE OF CLOSED-CYCLE POWER PLANTS OF DIFFERENT OUTPUTS (60 CYCLES PER SEC)

Net output.....kw	6000	12,000	25,000	50,000	100,000
Max pressure.....psia	600	600	850	850	850
Max diam:					
H-P turbine.....in.	17	19	26	35	47
L-P turbine.....in.	31	39	50	70	87
Max diam of axial-h-p compressor..in.	10	12	16	20	27
Max diam of axial-l-p compressor...in.	21	30	33	53	65

turbine, according to whether it is an axial- or radial-compressor type, and according to the pressure that is employed. For the greater part, however, the sizes of the machines will not prove very different from the figures indicated in Table 1. The fact that no regulating or stop valves are fitted to the machines leads to very favorable conditions for the construction of such hot-air turbines.

Such characteristics possessed by turbines permit the adoption of constructional forms that differ considerably from those usually employed for steam turbines. Special attention has to be paid to the inlet and outlet losses which, as a result of the small pressure and temperature drops that are utilized, play a relatively important part. The absence of any regulating devices permits of the turbine being situated, literally speaking, immediately in the piping that is conducting the working medium, an arrangement with which one is familiar in the construction of hydraulic machines. It is possible, for example, to pass the working medium through a bend-shaped supply pipe to the first runner wheel without employing annular channels.

The outlet can be made symmetrical and a considerable part of the kinetic energy can be recovered, for instance, in spiral outlet casings as in hydraulic turbines or pumps. Such forms are also used for the axial-flow compressors.

The small turbines also permit the adoption of double casings, which are designed according to the same principle as are the hot-air pipings referred to later. Between the thin internal shell and the external casing there is a layer of insulating material; the internal casing serves only for conducting the hot current, whereas the cold external casing takes up the pressure. The whole guide apparatus is fixed to the casing at a point where temperatures are low. In this manner, the external casing is, even for high inlet temperatures, subject at the most only to the discharge temperature of the last stage (approximately 925° F). Thus, although the turbine inlet temperature is much higher, heat-resisting steel need not be adopted for the casing, but only for the small internal parts.

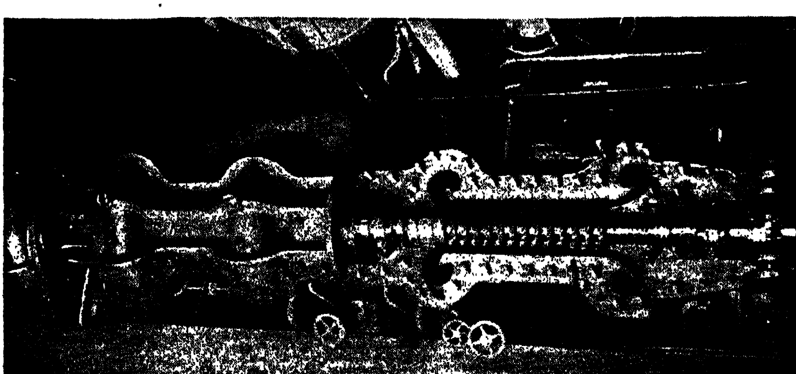
Similar constructional principles, dictated by the latest knowledge in the field of flow technics, apply also for the

compressor and for the intermediate coolers forming part of the latter.

Since relatively large quantities of air, but small increases in pressure, have to be dealt with, up-to-date axial-flow turbo-compressors of the multi-stage type are particularly suitable for such duty. The high speeds needed for this type of compressor lie within limits that offer good constructional conditions for the driving turbine also. As a result of these high speeds, the compressor set is of small dimensions even when dealing with the largest volumes.

It has been possible to raise the efficiency of the bladings above the values attainable with radial compressors. Again, this is the outcome of systematic research in this particular field (Fig. 1). Since turbine and compressor always and at all loads work at the same operating point, high-quality blading need only be developed for this one point without compromises. For these conditions, the course of the pressure-volume characteristic for part loads need not be considered. For the same reasons, for instance, in the case of small outputs or other gases, radial compressors can also be adopted, thereby leading to fewer stages.

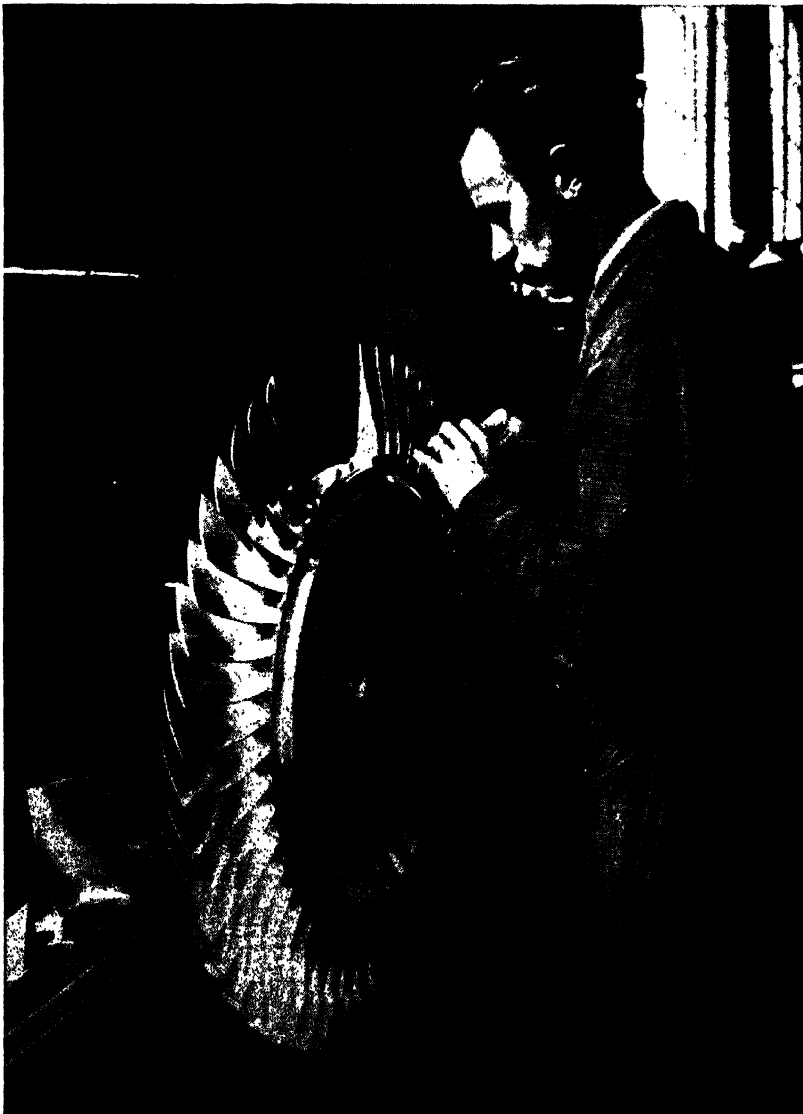
Figures 2 and 3 illustrate up-to-date blading of turbines and compressors having stage efficiencies of more than 90 per



Courtesy of Escher Wyss Engineering Works

Fig. 1. View of Compressor at Escher Wyss Plant with the Top Half Removed from One Unit.

cent. As a consequence of the raised pressure, the Reynolds' numbers of the machine bladings are of a considerably higher order, so that the percentage of friction losses becomes smaller.



Courtesy of Escher Wyss Engineering Works.

Fig. 2. Latest Type of Blading on Escher Wyss Turbine.

This holds good only for smooth and clean surfaces. Tests in Escher Wyss laboratories on a full-size axial compressor inhaling ambient air that contains only usual workshop impurities have proved that the blade efficiency dropped from 86 to 83 per cent during 12-hr continuous operation.

The rotating shafts of the machines are sealed from the ambient air by means of labyrinth glands or a combined system of labyrinth glands with liquid sealing, depending on the size of the plant and the kind of gas employed. Good sealing is necessary in consideration of the losses, especially at smaller outputs or when employing special gases.

The glands illustrated in Figure 4 have proved their merits in the case of the experimental plant. Sealing air extracted from the circuit is passed through pipe to the labyrinth chambers. The pressure of this sealing air is at all loads always somewhat higher than the pressure inside the glands, corresponding to the point where it is bled from the circuit. In this way, it becomes impossible for hot air to escape. From the point where sealing air is introduced, another part branches off toward the exterior of the gland and flows into a collecting space which is connected to a point in the circuit where a somewhat lower pressure prevails.

Outside these air-sealed labyrinth glands, there is a ring

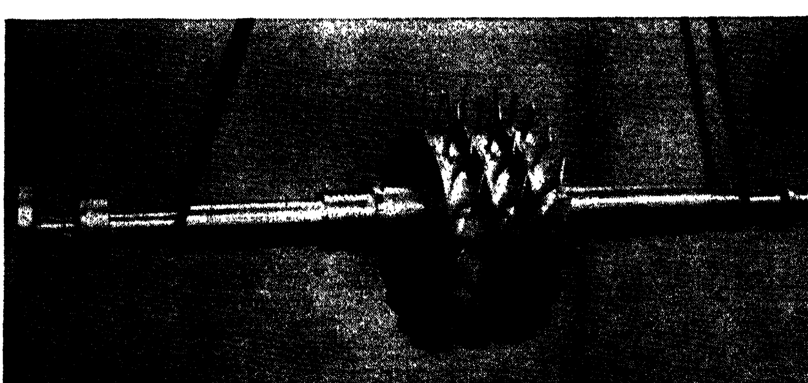


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Fig. 3. Latest Type of Blading on Escher Wyss Compressor.

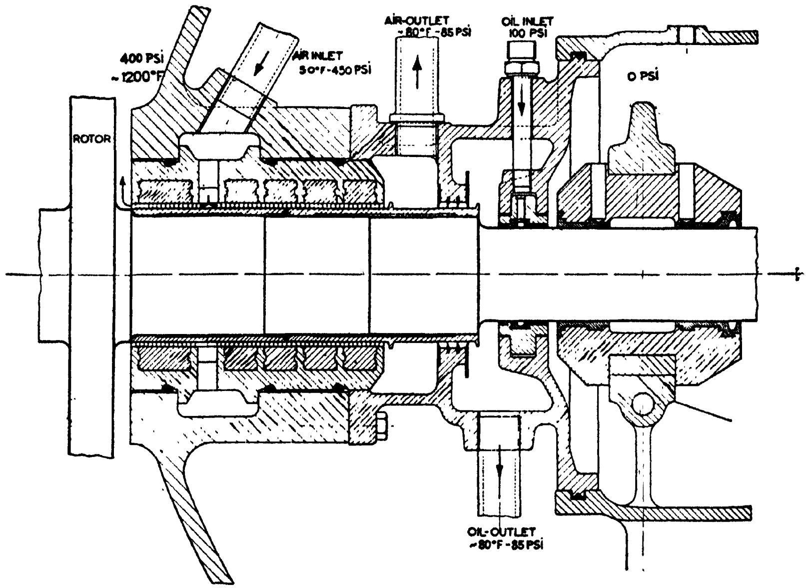

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Fig. 4. Turbine-bearing Construction Including Arrangement for Sealing and Cooling Air.

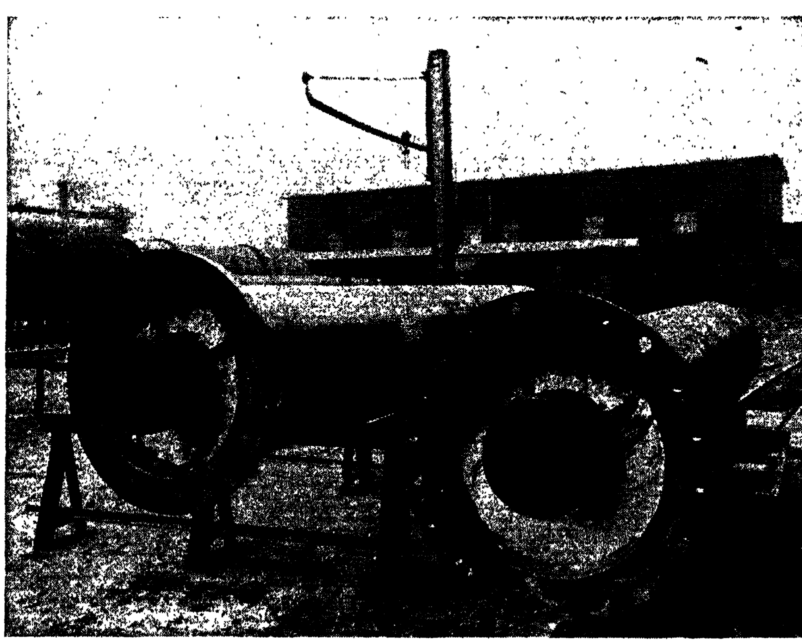

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Fig. 5. High-temperature Air Piping Showing Double-wall Construction.

through which oil under pressure is supplied to the shaft. This pressure oil prevents the escape of any air from the circuit. On either side of the pressure oil ring, means are provided for leading the oil to a reservoir, which is under a suitable pressure above atmospheric because it is connected to the interior of the circuit. The oil-sealing ring can also be combined with the bearing itself. By suitable connections of the sealing-air pipes to the circuit, it is possible to ensure that at all loads—that is, at all pressure levels—the sealing pressures will likewise rise and fall automatically and that the direction of the current will remain the same without regulating valves of any kind having to be interconnected. In practice, the foregoing is very important for ensuring the safety of the plant, and the measures described have proved fully satisfactory in trial operation under the most exacting conditions. The sealing air, introduced to the end sections of the hot turbine shafts, is simultaneously utilized in an advantageous manner for cooling these parts, so that the bearing sections remain quite cold.

Hot-air Piping

In order to reduce as far as possible the quantity of metal capable of withstanding high temperatures, the hot-air pipings have been executed with double walls according to the same principle as for the turbines (Fig. 5). The design comprises a thin-walled internal tube of a heat-resisting material serving only for conducting the stream of gas. This tube is relieved by bores from the pressure in the heat-insulating space which surrounds it. The heat-insulating space in its turn is enclosed by a thicker-walled pipe of standard material which can easily take up the pressure of the working medium, since it is protected by the insulating material and, therefore, is not under high temperature. The necessary measures are, of course, taken to prevent insulating material from gaining access to tube. With this design, much high-quality and expensive steel can be saved.

Air Heater

In the aerodynamic circuit, the air heater plays a similar part to that of the steam boiler in a steam turbine plant. The

heat of the fuel is imparted indirectly to the working medium by heat-transmission surfaces, the combustion gases being kept away entirely from the machines.

Since no feed water is employed, the air heater can, in principle and in contradistinction to a steam boiler, be installed in the open air without any building, all danger of freezing being nonexistent.

The design of the air heater is dependent on the fuel that has to be dealt with. Many forms are possible. Coal-fired air heaters can resemble in their design up-to-date steam boilers, as may be seen from Figure 6. According to the present stage of development, the space requirements of such coal-fired air heaters are not greater than those of up-to-date high-pressure

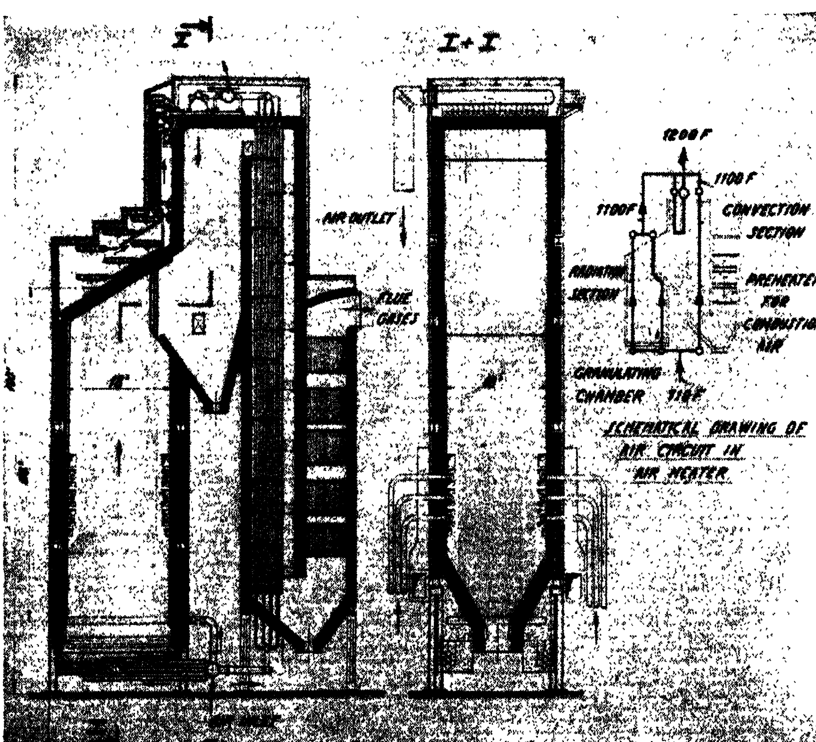

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Fig. 6. Coal-fired Air Heater for 12,000-kw Plant.

boilers. Figure 6 shows a section through a coal-fired air heater with granulating chamber. This project for a 12,000-kw plant is very conservative as regards the combustion chamber and

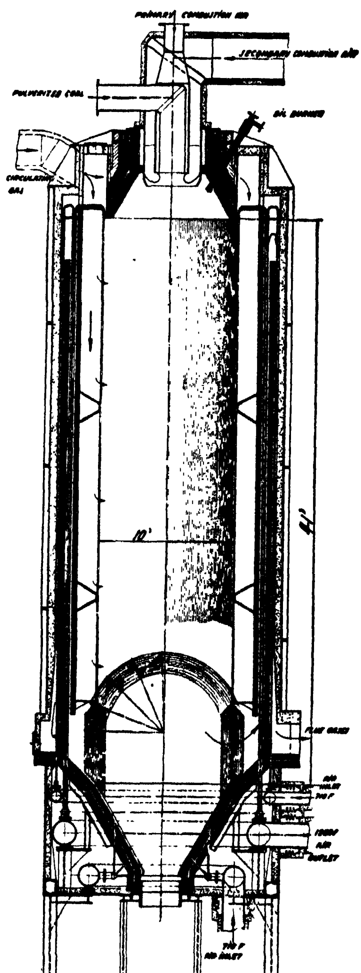

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Fig. 7. Air Heater for 6000-kw Plant Showing One Design for Pulverized Coal

the stresses and temperatures for the tube walls, since care in this connection appears necessary for a first execution. It may, however, be definitely expected that subsequent developments will lead to the heating surfaces and the dimensions being considerably reduced. The inlet temperature of the tube nest in

the convection section amounts to only about 1850° F. By returning the flue gases in the combustion chamber, the temperature of the combustion chamber is regulated and reduced. The tubes have diameters of approximately 1½ to ¾ in. with wall thickness of 0.1 to 0.15 in.

Figure 7 shows that a pulverized-coal-fired air heater can also be built favorably in a cylindric form, permitting the use of many equal elements or tube bundles. By blowing the circulating return gases back again into the combustion room all along the inner tube wall in the space between the tubes, the slag and ash particles can be chilled down and the tube surface kept clean. The space needed is about the same as for an oil- or gas-fired heater. The use of preheated combustion air leads to an increase in the furnace temperature, particularly in cases where pulverized coal is used. For example, in plants burning pulverized coal, such preheating of the secondary air is desirable, since it permits a reduction in the size of the combustion chamber.

Figure 8 shows an example of an oil-fired air heater for a 6000-kw plant with 600 psia and double heating. The tubes are arranged around a supercharged combustion chamber. The tube walls for heating the air in the first and second stages, 600 and 200 psia respectively, are united in a common heater. In this project, the combustion chamber is lined with refractory material, and the heat is given up to the tubes mainly by convection. By raising the pressure of the combustion gas to about 45 to 70 psia and with velocities of combustion gases from 100 to 160 ft per sec and air velocities inside the tubes from 65 to 130 ft per sec, the heating surface can be kept small. For such projects, it amounts to only 0.5 sq ft per kw net output of the plant. The total weight of iron in such an air heater is less than nine pounds per kilowatt. Of this figure, the proportion of alloyed steels is about 50 per cent. The tubes, which may expand in operation by about two inches, can move freely in the upward direction. In order to keep the hot-air piping as short as possible, the heated air for the high- and low-pressure turbine is discharged below. The shell of the heater is of ordinary steel and made airtight, and can be dismantled in a num-

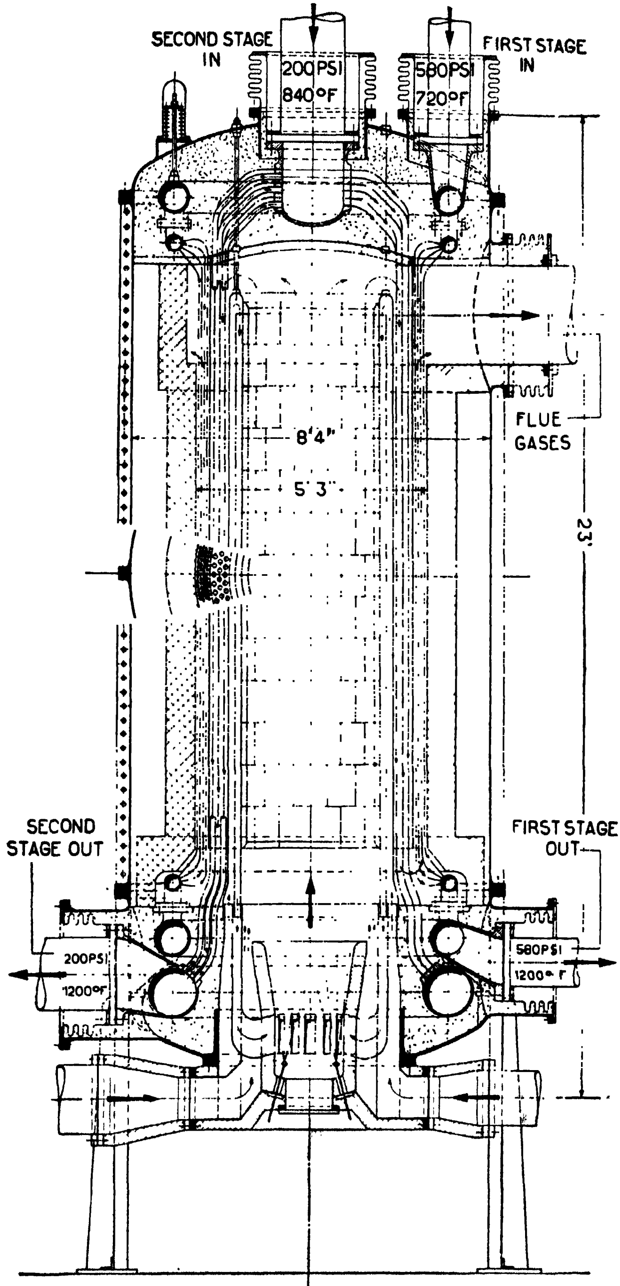

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Fig. 8. Oil-fired Air Heater for 6000-kw Plant Using Double Heating.

ber of sections, so that the tubes are easily accessible from the side and can be removed without difficulty. At the same time, the shell serves as a support for the tubes.

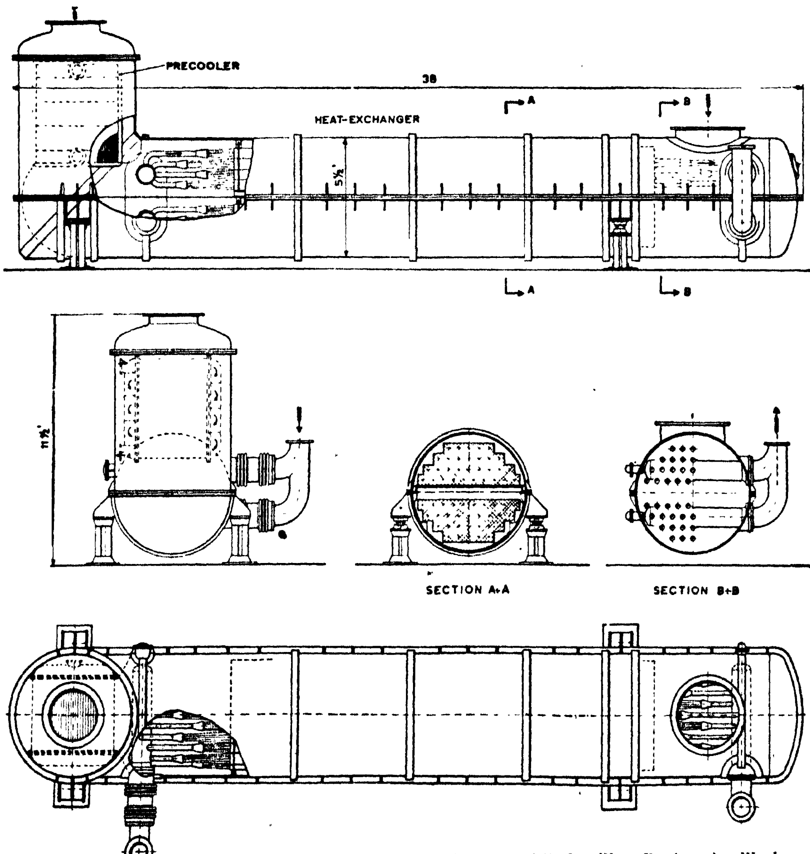
Since the waste gases are still of high temperature (approximately 1000° F), they are expanded in an exhaust turbine and simultaneously cooled. The exhaust turbine drives the compressor for the combustion air. In this way, a preheater for the combustion air can be entirely eliminated. The supercharging set need not be particularly efficient because most of the losses are recovered in the firing. Since there is no regenerator, there is little danger of detrimental soiling. The design of an air heater employing blast-furnace or natural gas is, for the greater part, the same as for oil firing. The heating surfaces are likewise very similar.

Heat Exchanger

Figure 9 illustrates the design of a tubular exchanger operating according to the countercurrent principle and transmitting a maximum amount of heat with a minimum loss of pressure. The high-pressure air flows through the interior of the tubes, and the external shell has only to withstand the lower back pressure. The apparatus can be arranged as desired according to the available space, or it can be subdivided into two or more parts. The use of normal tubes of small diameter permits standardization of all heat-exchanger elements for the various outputs, and thus, to a large degree, permits manufacture in series.

Tubes of only 0.15 to 0.25 in. in diameter are used in the plant under consideration. They are spaced by special distance pieces that offer little resistance to the current of air. A large number of these thin tubes is united in a tube nest. In turn, the tubes of the nest are connected by a small number of collector pipes. Separate removal of each tube nest is easily possible. In the case of stationary plants, this can be effected, for example, by drawing the various tube sections out of the heat exchanger in an axial direction. For marine installations, the shell of the heat exchanger is split and can be easily lifted, thus giving access to the tube nests from above. The tightness

of each nest can be checked separately. However, since a perfectly clean medium flows on either side of the tube walls, there is no reason to fear interruptions in the operation. Furthermore, the highest temperature in the heat exchanger



Courtesy of Escher Wyss Engineering Works

Fig. 9. Countercurrent Tubular-type Heat Exchanger.

amounts only to about 850° F, so that it is unnecessary to employ special-quality metals for the tubes.

The temperature of the low-pressure air, on issuing from the heat exchanger, is about 200 to 250° F, after which it passes to a precooler through which water circulates for cooling as far as possible—that is, down to the inlet temperature of the compressor, the object being to reduce the compression work.

Since, as a result of the raised pressure, the heat-transmission coefficients on the air side are also favorable in the precooler and in the intermediate coolers of the compressor, the surfaces and dimensions of these apparatuses are not large, in contradistinction to water-cooled air coolers and intermediate coolers which are of standard ribbed-tube design with water circulation through the tubes.

In many cases, it is advantageous to combine the low-pressure turbine arranged in the circuit before the heat exchanger,

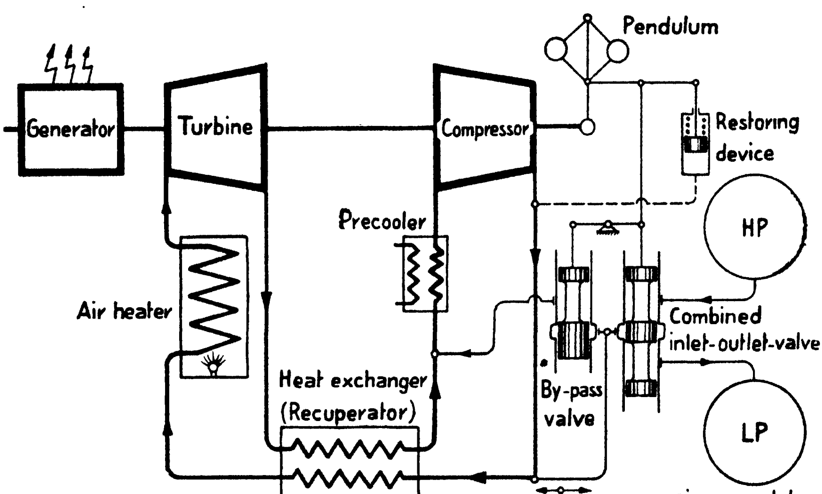


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Fig. 10. Regulator Diagram of Simplified Type of Closed-cycle Compressor Turbine Unit.

in the manner illustrated in Figure 10. In this way, additional pressure losses are avoided and the stream of air passes directly to the tube nest. As may be seen from Figure 10, the heat exchanger, whether arranged in this or some similar manner, is an exceptionally simple apparatus that operates very reliably.

Governing

For raising the output, air is introduced to the circuit from a high-pressure accumulator of cold air; for reducing the output, air is withdrawn from the circuit and passed to a low-pressure accumulator. Automatic governing for an installation having

rigid couplings between the machines takes place fundamentally as follows: when the load is thrown off, the consequent rise in speed influences the centrifugal governor (pendulum) which causes the discharge side of the combined inlet-outlet valve to open (Fig. 11) so that air from the high-pressure branch of the circuit issues into the low-pressure accumulator *LP*. On the other hand, when the load is thrown on the plant, the resulting drop in speed causes the inlet side of valve to open, as a consequence of which air from the high-pressure ac-

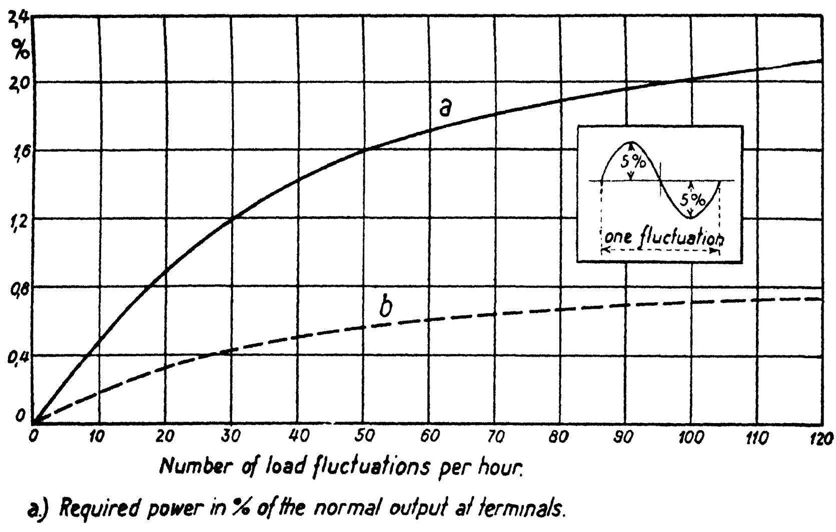

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Fig. 11. Power Consumption for Regulation of Pressure Level.

cumulator *HP* is admitted to the circuit at the same point. For small reductions in load and, consequently, for only slight increases in speed, the main valve operates only within a small range without opening either the inlet or outlet. On the other hand, the by-pass valve opens and by-passes air from the high-pressure side of the circuit to the low-pressure side without developing output, so that the useful output of the plant is reduced. By regulating small and frequently recurring load fluctuations with the by-pass valve, the air consumption from accumulator *HP* for purposes of regulation is reduced, and charging work is thus saved.

Regulation by supplying or withdrawing working medium to or from the high-pressure side of the circuit has the advantage of immediate efficiency, since the pressure ratio PH/PL is immediately raised on air being admitted, thus causing the plant to give up additional output. Inversely, when air is with-

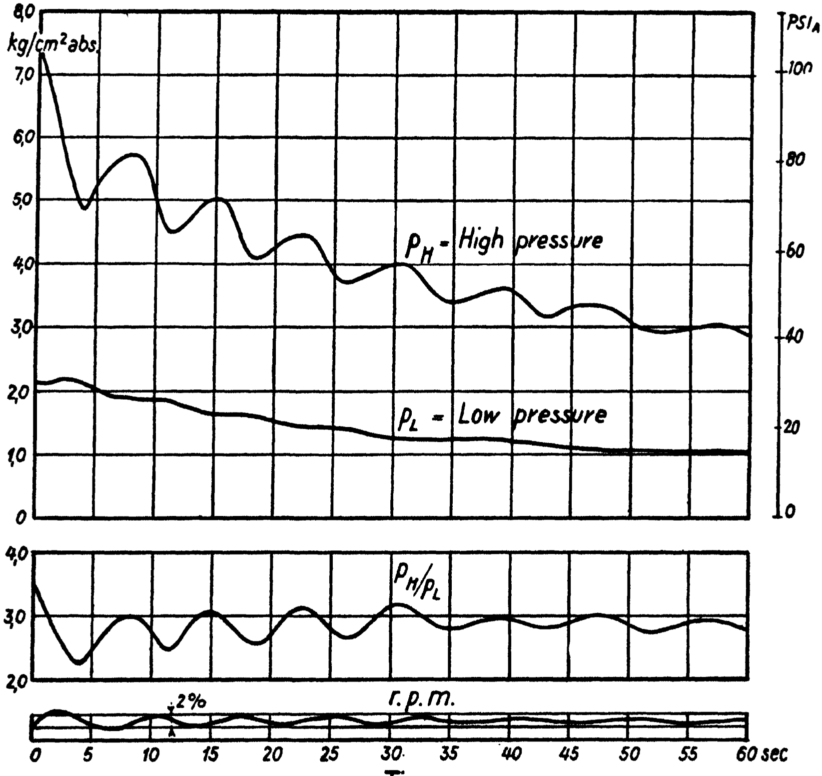

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Fig. 12. Curves of Pressure and Speed During a Switch Off.

drawn from the high-pressure side, the pressure ratio immediately drops, so that, for example, when suddenly withdrawing less than 20 per cent of the air content of the circuit, it has dropped to such an extent that the transition from full to no load has already taken place. On the other hand, this *momentary effect* would be unsuitable for the supply or withdrawal of working medium on the low-pressure side.

When admitting or withdrawing air on the high-pressure side of the circuit, the consumption of air for such regulating purposes is, in the case of quickly recurring, small, periodical load fluctuations, no longer proportional to the number of actual load fluctuations. Instead, consumption of air increases relatively less, since insufficient time remains between the separate fluctuations for reestablishing the stationary pressure

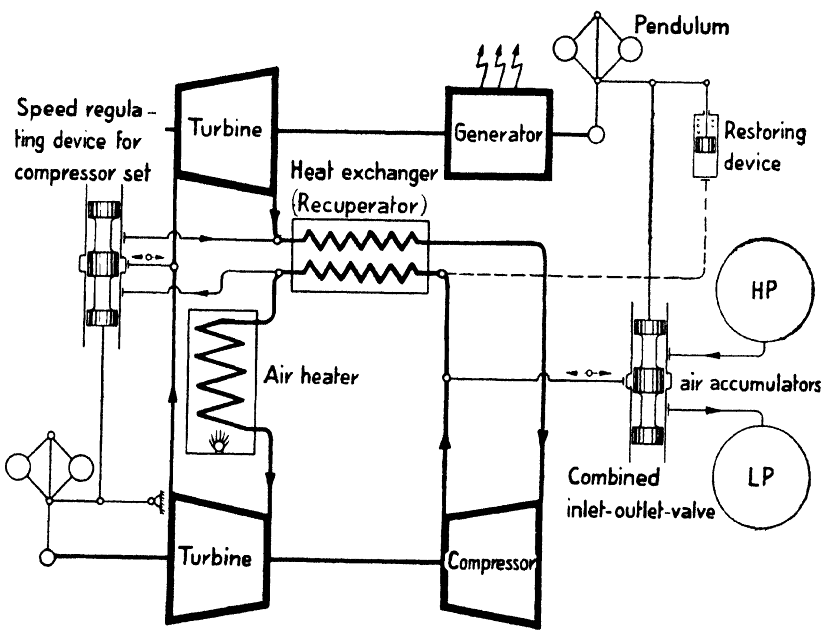

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Fig. 13. Regulator Diagram Where Two Separate Shafts Are Used in the Closed Cycle.

ratio, and the control consequently takes place chiefly under the influence of the momentary effect (Fig. 12).

The influence of the momentary effect on the governing, which for known machine characteristics can also be calculated theoretically, has been checked by experiments. The pressure course (Fig. 13) plotted for a load-reducing action shows that, during the first moment, the equilibrium in output is brought about by reducing the pressure ratio PH/PL, and that the pressure level only gradually drops to the final condition.

The regulation of the furnace, not indicated on the schematical drawing of Figure 11, need not take place very quickly, thanks to the accumulation of heat in the heaters and apparatuses. Changes in the power output of the turbine (due to smaller deviations of the final heating temperature of the working air from its stationary value) are compensated by automatic and temporary raising or lowering of the pressure level by a small amount.

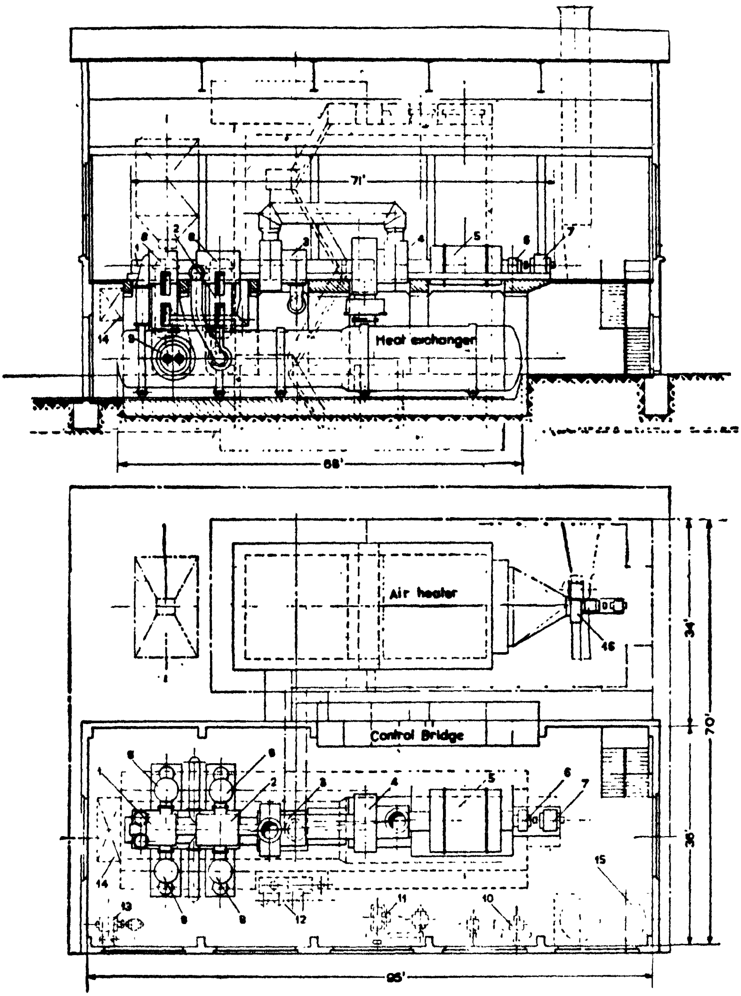
The method of regulation herein described is advantageous because the regulating means can be accommodated in regulating valves outside the circuit proper; because these means are traversed only by cold air; and because, under normal working conditions, the air of the circuit does not pass through the regulating means, so that no additional throttling losses result.

For installations in which the turbine developing useful output is separated mechanically from the compressor set, which is particularly the case where useful output has to be given up at different speeds (for example, compressor drive and ship propulsion), the compressor set operates in the normal operating condition, which remains stable and does not require any special regulation. The equilibrium is disturbed during the supply and withdrawal of working medium for changing the useful output. During this period, deviations from a given speed range for the free-running compressor set are prevented by partly by-passing either the turbine that is developing useful output, or the turbine driving the compressor, by a special valve which is influenced from a speed-limiting governor of the compressor set (Fig. 13).

The test plant of Figure 13 has worked for long periods entirely separated from the municipal network and has supplied the whole works of Escher Wyss. On these occasions, regulating governing, in its simplest form, proved satisfactory.

Projects for Power Stations

There are no special requirements for the disposition of the various machines and apparatuses of a closed-cycle plant



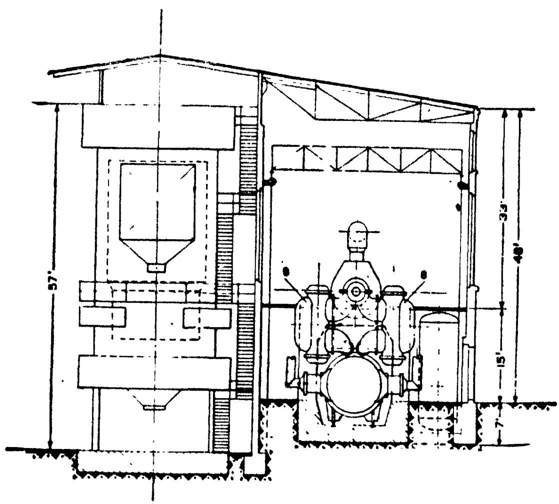
Courtesy of Escher Wyss Engineering Works

Fig. 14. 12,000-kw Complete Plant with the Closed Cycle. The Air Heater with Pulverized Coal Burner All in the Open.

within the available space. Air is not subject to the force of gravity like steam condensate, so that differences in level, such as are required for ensuring passage of the condensate and feed water, need not be provided for. Thus, the machines and apparatuses can be combined in a small, closed set having short connecting pipes. Such a combination reduces losses of pres-

sures and temperature to a minimum, which is of the utmost importance in such plants.

Figure 14 illustrates a 12,000-kw plant with pulverized coal firing for industrial purposes. The air heater is installed in the open, next to the machine house, thus considerably reducing the cost of the building. The plant is of the single-stage ex-



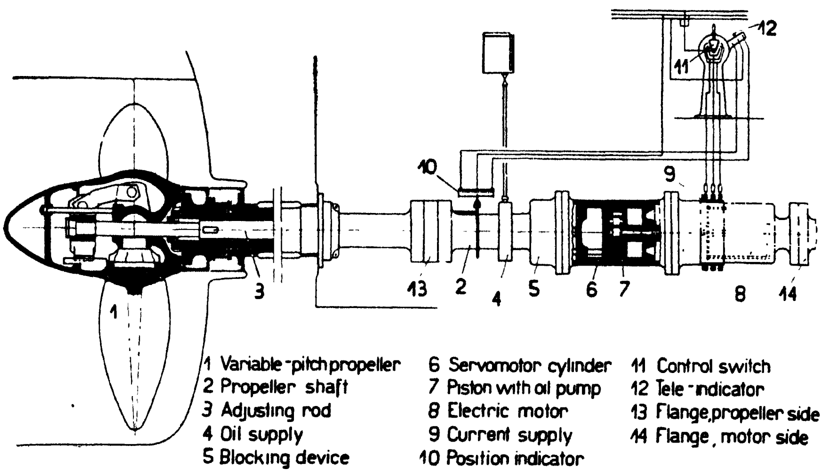
Legend	
1	LP compressor
2	HP compressor
3	HP turbine
4	LP turbine
5	Turbo-alternator
6	Exciter
7	Starting motor
8	Intercooler
9	Precooler
10	Loading compressor for air accumulator
11	Loading compressor for circuit
12	Regulating station
13	Cooling-water-pump
14	Oil-tank
15	Air accumulators
16	Exhaust-gas blower

Courtesy of Escher Wyss Engineering Works

Fig. 14 (Cont.). 12,000-kw Complete Plant with the Closed Cycle.

pansion type. The working pressure at full load amounts to 400 psia before the turbine; the maximum temperature, to 1200° F; and the back pressure, to 115 psia. The machines are all arranged in one row. The high-pressure turbine drives the compressor serving the circuit and the low-pressure turbine drives the generator. The heat exchanger is installed underneath the machine set but can also be arranged elsewhere (for

example, in the open) depending on the available space. The weight of the circulating air is about 250 lb per sec. All the chief auxiliary drives that are required may be seen in Figure 14. Their small number in comparison to the auxiliaries of up-to-date steam-power plants is characteristic. The over-all thermal efficiency of such an installation amounts to at least



Courtesy of Escher Wyss Engineering Works

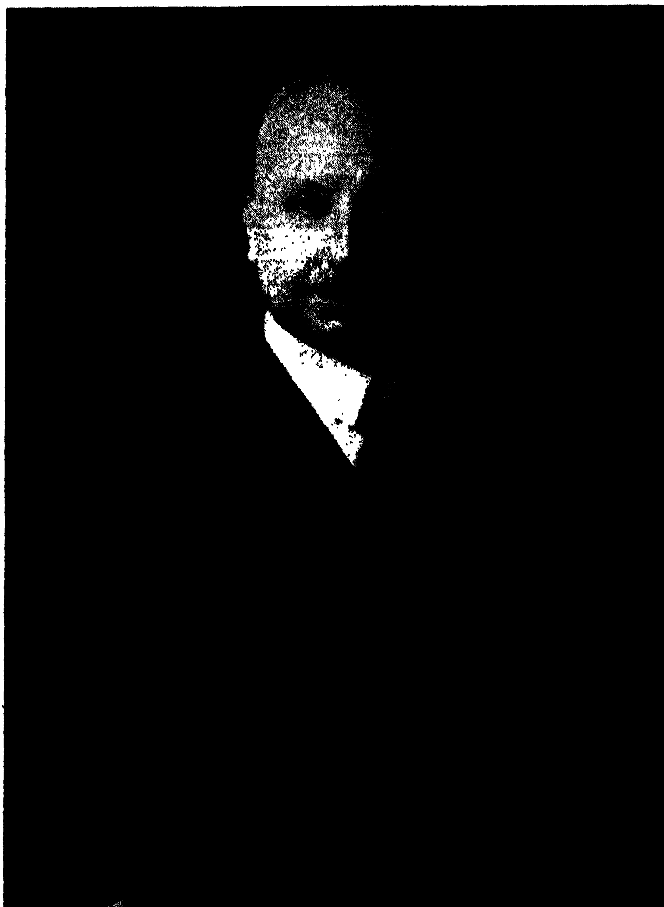
Fig. 15. Reversible Propeller Designed for Gas Turbine Marine Plants. This propeller not only eliminates the use of a reversing turbine but is much more flexible and efficient.

32 to 33 per cent at full load, 31 per cent at half load, and 28 per cent at quarter load. Naturally, these figures vary somewhat according to the quality of the coal.

Escher Wyss has a number of reversable pitch propellers in service as shown in Figure 15. These are ideally suited to gas turbine driven ships using either the open or closed cycle system.

PART II

TURBOCHARGERS



Dr. Alfred J. Buchi, the Father of Modern Low- and High-pressure Turbocharging Systems.

Chapter VIII

American Locomotive Company

The Alco six-cylinder $12\frac{1}{2}$ by 13-in. engine is turbocharged on the Buchi system, as shown in Figure 1. The turbocharger installation was especially developed for locomotive service, where rugged construction and reliable operation are of utmost importance. Certain alterations have been made in the standard 660-hp Alco Diesel engine to adapt it to the use of the turbocharger. These alterations include changes in the combustion chamber, fuel-injection system, and valve timing. The air and exhaust manifolds have also been modified to suit the turbocharger. The exhaust muffler has been removed from its position above the generator, and the turbocharger put in its place. It has been found unnecessary to use a muffler with the turbocharged engine.

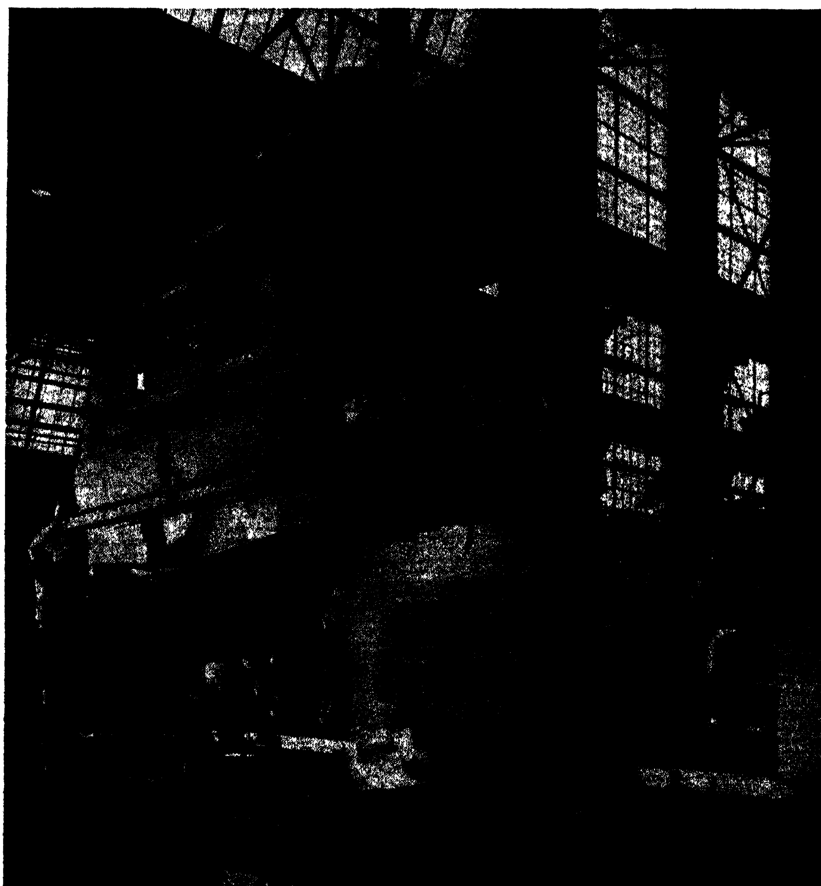
After exhaustive tests of the turbocharged engine, a conservative rating of 1000 hp has been established as standard without materially adding to the bearing pressure, but with better fuel economy and lower running temperatures. The greater volume of air used in scavenging the engine lowers the temperature of the cylinders, making it possible to maintain a better oil film on the cylinder walls. A phantom view of this turbocharger is shown in Figure 2.

Principle of Operation

The operating cycle of the engine is the same as a nonsupercharged engine with the exception of the exhaust and suction strokes of the cycle.

The camshaft of the engine is designed to allow the opening period of the air and exhaust valves to overlap to a greater extent than in the nonsupercharged four-cycle engine, permit-

ting the turbocharged air simultaneously to clean out the exhaust gases in the cylinder near the end of the exhaust stroke and to cool the combustion chamber of the engine. For complete scavenging, a much lower pressure in the exhaust manifold than is present in the air manifold is necessary to insure sufficient flow of scavenging air through the combustion chamber. Low pressure is attained by providing two exhaust manifolds, each being supplied by three cylinders, which conduct the exhaust gases into the exhaust-gas turbine casing at two

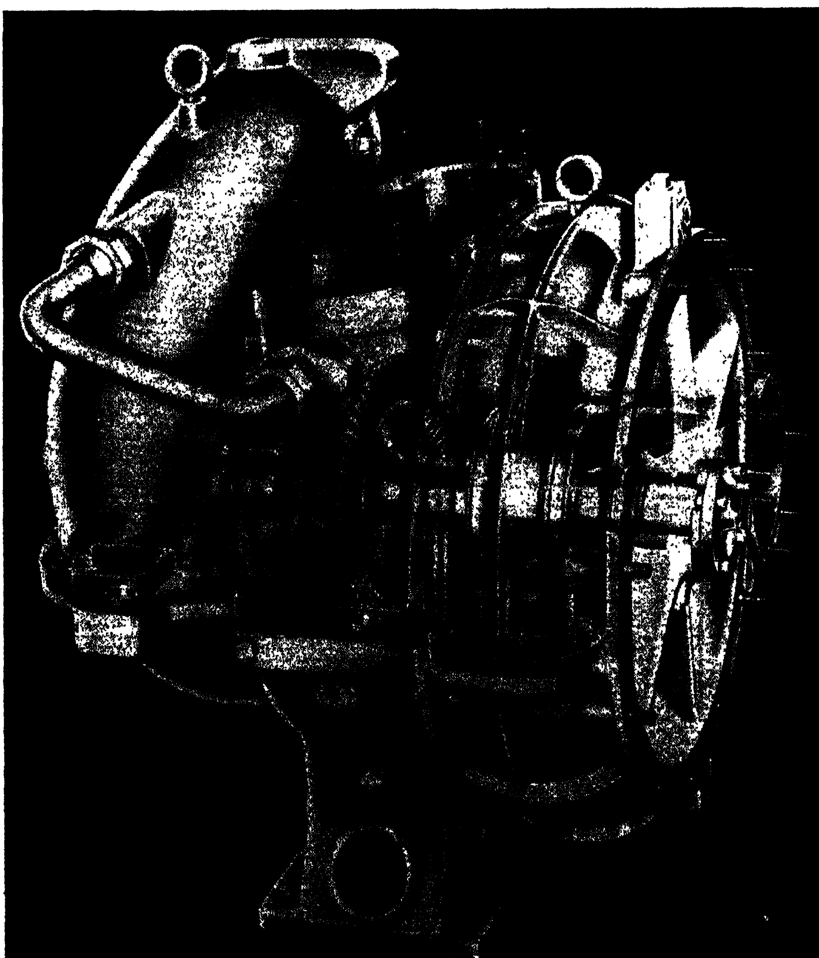


Courtesy of American Locomotive Company

Fig. 1. Alco Power Unit Being Lowered on to Locomotive Frame. Note turbocharger mounted above generator.

separate points. By doing this, it is possible to have a pulsating pressure of a minimum value much lower than the air-inlet pressure during the period of overlapping of the air and exhaust valves. This permits the clean cool air to scavenge the combustion chamber thoroughly.

The scavenging system described above makes it possible to trap a much larger quantity of clean air in the combustion



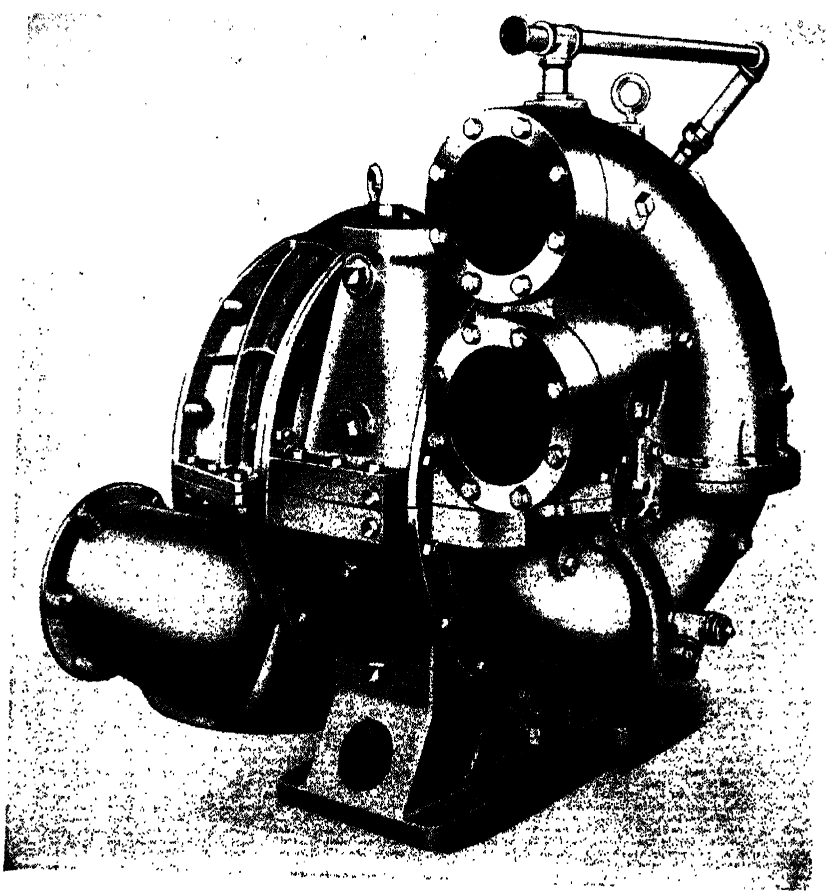
Courtesy of American Locomotive Company

Fig. 2. Phantom View of Alco Buchi Turbocharger Showing Turbine Wheel on the Left and Blower on the Right.

chamber at the beginning of the compression stroke. A larger quantity of fuel oil may be burned, thereby taking fuller advantage of the engine's volumetric capacity than is possible on the nonsupercharged four-cycle Diesel engine.

Description of the Turbocharger

The turbocharger consists of a centrifugal blower driven by an exhaust gas turbine (Figs. 3 and 4). Both blower and turbine are mounted on a common rotor shaft and separated by a water-cooled diaphragm. Labyrinth glands are provided

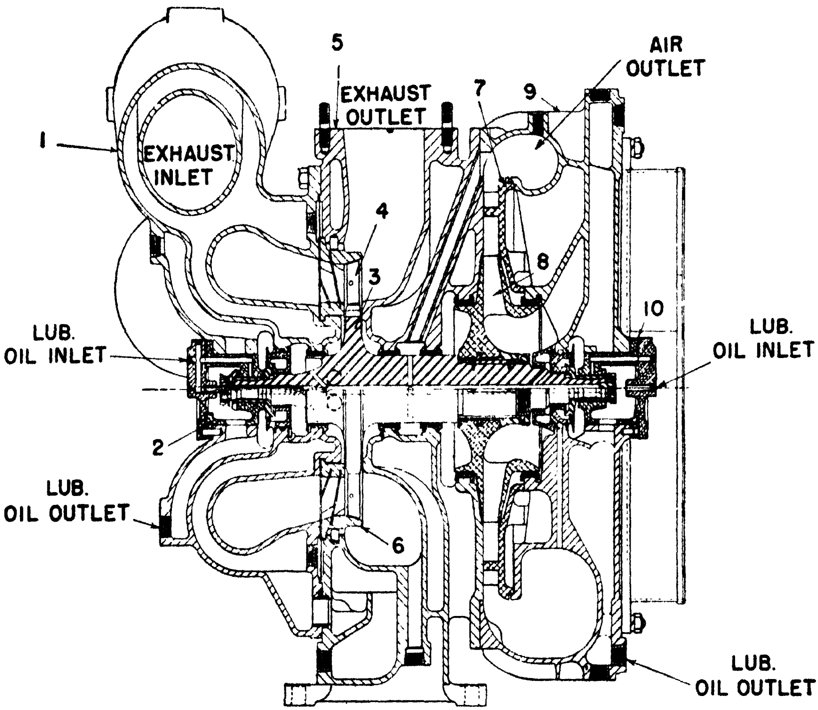


Courtesy of American Locomotive Company

Fig. 3. Outside View of Turbocharger.

to seal the individual suction and discharge chambers to prevent contamination and pressure losses.

The speed of the turbocharger is dependent upon the volume of exhaust gases produced by the engine. Consequently, the turbocharger is directly controlled by the load on the en-



Courtesy of American Locomotive Company

1—Exhaust Inlet Casing
2—Turbine End Bearing
3—Rotor Shaft
4—Turbine
5—Intermediate Casing
6—Turbine Nozzle Ring
7—Blower Diffuser
8—Blower Impeller
9—Blower Casing
10—Blower End Bearing

Fig. 4. Cross Section of Turbocharger.

gine, and the automatic control of the turbocharger is thereby provided.

The blower casing is divided at the center line of the shaft into upper and lower halves. The casing is bolted securely to the other parts of the turbocharger housing. One of the two rotor shaft bearings is carried in a housing mounted in the

blower casing. The impeller is housed in a suitably shaped space in the blower casing, and the outer rim of the impeller is surrounded by the stationary bladed diffuser. Axially converging entrance passages surround the bearing housing. The air flows from these passages into the impeller and is discharged radially from the impeller to the diffuser. The blades in the diffuser are shaped to decrease the velocity of the air and at the same time to increase its pressure. The air discharged radially from the diffuser is collected in a spiral chamber and delivered to the air manifold of the engine through the discharge flange. This flange and the spiral chamber are part of the blower casing.

Exhaust Turbine

The central portion of the turbocharger is made up of two castings, with the dividing line on the horizontal-shaft center line. The lower half has an integrally cast foot for mounting the entire assembly on its foundation. A large opening in the upper half provides the connection to the exhaust stack. On the exhaust side, a nozzle ring is mounted which serves to direct the exhaust gases against the turbine blades at the proper angle.

The turbine blades rotate within the rim of the nozzle ring. Gas from the nozzle ring flows across the blades and leaves the blades in an axial direction. The collection and discharge chamber, which is part of this central portion, is provided with water-cooled walls throughout.

The turbine consists of a forged-steel, one-piece shaft, mounted in bearings at each end. The air impeller, described above, is fitted near one end of the shaft. The turbine rotor is a large, integral flange, located near the other end of the shaft in which the turbine blades are fitted and locked. A lashing wire passes through a hole in each blade and is silver-soldered in place. The wire is made in several lengths so that groups of three, four, or five blades are independent of adjacent groups.

The turbine nozzle ring is divided at the horizontal center line, and has a flange on the exhaust-inlet-casing side which fits into a counter-bore in the intermediate casing. There is a

small clearance between this flange and the exhaust casing. There are two separate nozzles in this assembly which are independently connected to the upper and lower exhaust manifolds, respectively, through passages in the exhaust inlet casing. In each of the nozzle ports, there are eight cast-in steel vanes.

Exhaust Inlet Casing

This casing is provided with suitable passages for conducting the exhaust gases from the manifolds to the nozzle ring. It is made of cast iron and is water-jacketed. In assembly, the casing halves are bolted together at the horizontal center line and to the intermediate casing. The exhaust end-shaft bearing is carried in a housing mounted in this casing.

Turbocharger Air Filter

To prevent contamination of the air delivered to the engine, the blower intake is fitted with a filter. The filter consists of elements saturated with oil to which any foreign elements fine enough to get into the filter will adhere. To insure the proper intake volume and cleanliness of the blower passages, the filter must be cleaned at regular intervals, depending upon the type of service and operating conditions.

The filter should be removed from the engine and the elements thoroughly washed with kerosene or fuel oil, and then saturated with SAE 20 or 30 lubricating oil.

Bearings

The rotor is carried at both ends by sleeve bearings. The bearing at the turbine end is a combined supporting-and-thrust bearing for fixing the axial position of the shaft. The bearing at the blower end is arranged to allow the free heat expansion of the shaft. The journals are hardened-steel sleeves, which may be replaced readily.

Labyrinth glands of nickel or aluminum are provided throughout to prevent the contamination of the gases and bearings. The bronze glands, which seal the shaft areas, are

machined in the form of a bushing, and thus make replacement a simple matter.

TABLE 1

ADJUSTMENT DATA

Volume of indrawn air, measured in suction branch....	3000 cfm
Suction temperature, measured in suction branch......	84° F
Suction pressure, measured in suction branch[1].........	−0.32 psi approx
Charging pressure, measured in delivery branch[1].......	5 psi approx
Approx blower speed at full load....................	10,300 rpm
Approx blower speed at engine idling speed...........	2200 rpm
Max blower speed................................	13,000 rpm
Max pressure of exhaust gases after turbine[1]..........	−0.2 psi
Approx exhaust-gas temperature after exhaust valves..	850° F
Approx exhaust-gas temperature before turbine........	950° F
Max allowable temperature before turbine............	1050° F continuous 1100° F momentary
CLEARANCES	
Bearings:	
running..	0.006 to 0.008 in.
thrust...	0.004 to 0.006 in.
Shaft:	
radial clearance	
labyrinths, turbine end bearings........	0.009 to 0.011 in.
three center labyrinths..........	0.009 to 0.011 in.
two blower hub labyrinths...........	0.009 to 0.011 in.
oil slinger............................	0.015 to 0.020 in.
Blower impeller:	
axial clearance between impeller and center housing..	1/16 in.
Exhaust turbine impeller:	
radial clearance................................	0.018 to 0.022 in.

[1] Gauge pressure

Operation and Maintenance

On inspection, check the following:

1. The blower speed. (The maximum allowable speed is marked on the turbocharger name plate.)

2. The sections of the air intake filter are to be cleaned every week, or as needed.

3. Check the air pressure at full load. For the proper temperatures, pressures, and speeds, see *Adjustment Data,* Table 1.

4. The free running time of the rotor, after stopping the Diesel engine, should be checked from time to time. This is

an indication of the condition of the bearings. The free running time from 2200 rpm (engine idling) to a standstill is approximately 2.8 to 3 min.

If the blower speed or the charging air pressure drops under the normal values, or if the temperature of the inlet exhaust gases at normal load with pressure charging, exceeds the maximum allowable temperature for continuous running, immediately reduce the engine speed and ascertain the cause of the trouble, which may be due to:

1. Failure of the fuel-injection system or other trouble with the Diesel engine—that is, leaky exhaust valves, excessive piston blow-by, and so forth.
2. Losses from leaky joints in the air-delivery piping.
3. Losses in the exhaust-gas piping between the Diesel engine and the turbine.
4. Excessive restriction in the air filter.
5. Troubles with the blower assembly. (Bearing troubles, rubbing of packing gland or turbine wheels, and so forth.)
6. Or look for troubles elsewhere, such as a poor connection in the generator field resistance, slipping of exciter belts, and so forth.

If the turbocharger vibrates abnormally, the cause should be found and rectified immediately. The vibration might be caused by a disturbance of the rotor balance combined with too big a clearance in the bearings. A slight bend in the shaft or variation from normal of a similar nature may be responsible.

Lubrication

The turbocharger bearings are lubricated by oil taken directly from the engine lubricating system.

Packing glands are provided around the inlet pipes at the turbocharger bearing covers. The oil is drained from each bearing into the engine and recirculated through the system.

Dismantling

The turbocharger should be dismantled only by a competent mechanic.

Before attempting to dismantle the turbocharger, the cooling water is to be drained and all the lubricating-oil piping is to be dismantled. The two exhaust-pipe connections must be disconnected and the stack connection removed. Clear space above the turbocharger is essential. The turbocharger should be removed from the locomotive for disassembly. The two shaft end covers may now be taken off. All the bolts along the joint on the horizontal center line are taken out. The three sections of the upper casing may now be removed as one unit by lifting straight upward with the two eyebolts provided for this purpose. This operation will expose the turbine wheel and blower impeller, all the upper labyrinths, diffuser, and nozzle ring for inspection. Most inspections and repairs can be made from this condition.

To remove the turbine end bearing, it is necessary to remove the shaft end nut, lockwasher, and thrust washer. The bearing may then be gently removed axially. If done easily, the shaft will sink onto the labyrinths without becoming damaged.

To remove the blower end bearing, the shaft end nut and lock nut are taken off, and the bearing is slid out axially. If done smoothly and easily, the shaft will sink onto the labyrinths without becoming damaged. When both bearings are removed, the shaft and rotors can be lifted out of the lower casings.

In order to remove the nozzle ring and diffuser halves, it will be necessary to unbolt the outer casing halves on each side of the center section. The nozzle ring and diffuser can then be removed without further dismantling. Ordinarily, it will be unnecessary to remove either of these parts.

Inspection of Vital Parts

The blower should be dismantled at each annual inspection. The following points should be carefully checked:

Blower. Does the impeller fit firmly on the shaft and are there any signs of contact on the outer periphery or on the inlet rings?

Rotor Shaft. Are there any signs of contact caused by the labyrinths? Pitted running surfaces should be refaced. The surfaces under the labyrinth glands are to be examined at least once a year for corrosion.

Turbine Wheel. Are any of the blades broken or bent, and are the inlet edges still sharp? Have the blades become badly worn, or has the wheel been touching any point? Is there any sign that foreign matter has passed through the wheel? If blades are damaged in any way, the complete rotor should be returned to the manufacturer for rebalancing.

Nozzle Ring. Has the ring a firm fit or has it warped? Are there any signs of contact with the tips of the turbine wheel blades? Is there any foreign matter lodged between the nozzle plates, and have any plates worked loose, been burned, cracked, or bent? Bent or broken plates should be inspected by a qualified mechanic; the rotor should be rebalanced by the manufacturer.

Bearings. Is there any discoloration of the bearings caused by overheating? Are any defects visible, or do the running surfaces indicate that foreign particles have been rolled in them? Are the bearings receiving sufficient lubrication and are there any signs of excessive wear? The running clearances of the bearings should be checked to the proper figures given under *Adjustment Data,* Table 1.

Labyrinths. Check the labyrinths to see if any dirt has entered and see if the clearance has been affected by wear or corrosion by exhaust gases. Have any of the rings been bent or damaged in service or when assembling? The impeller labyrinth rings should be inspected to see if the sealing-air inlet is clean and correct.

Diffuser. The diffuser is to be checked for cleanliness, position, and general condition. The diffusers should be correctly fitted with reference to the direction of rotation of the impellers.

Casings. The turbine and impeller casings should be checked for any signs of leakage or large accumulations of dirt caused by oil, water, scale, or dust.

Reassembling

To reassemble, the procedure described under *Dismantling* should be reversed, care being taken in the handling of the rotor not to damage the labyrinths. After replacing the shaft and bearings, the thrust bearing clearance at the turbine end should be checked *without fail*. After the three upper casing halves have been replaced, the clearance between the turbine runner and the nozzle ring (accessible through the stack connection) *must* be checked.

The bearings are to be checked carefully at the annual inspection. If necessary, new bearings and sleeves are to be installed.

The neoprene seal rings at the grommets on the turbine side should be renewed annually or any other time should they be found in questionable condition, preferably each time the turbocharger is dismantled.

In replacing the parts, all clearances should be checked, all revolving parts should be running on true centers, and the turbine shaft should run freely. After the turbocharger is completely assembled, it should be tested to the temperatures and speeds given under *Adjustment Data.*

Note: A detailed theoretical discussion of the Buchi System is given in *The Modern Gas Turbine* (2nd Ed., 1947), by R. Tom Sawyer, published by Prentice-Hall, Inc.

Chapter IX

The Elliott Company

The Elliott Company manufactures three sizes of turbochargers. The small size, BF-26, has a maximum speed of 21,000 rpm and is suitable for Diesel engines of 400 to 700 hp. The medium size, BF-34, has a maximum speed of 16,000 rpm and is suitable for engines of 600 to 1200 hp. The large size, BF-44, is suitable for engines of 1000 to 1900 hp. The BF-44 has a maximum speed of 12,500 rpm. The above figures are based upon the use of one turbocharger on an engine. See Tables 1, 2, and 3.

These three turbochargers are very similar to one another in construction; consequently, they will be treated as one in this chapter. Figure 1 shows a cutaway of the turbocharger with its own lubricating-oil pump and inlet-air silencer. The same unit may be built to obtain its supply of lubricating oil from the engine system.

Figure 2 shows a typical installation of the turbocharger on a six-cylinder Worthington Diesel engine.

The Elliott turbocharger may be found in all classes of service in which the heavy-duty Diesel engine is now operating, which includes stationary; marine; railway; and heavy, portable Diesel power plants.

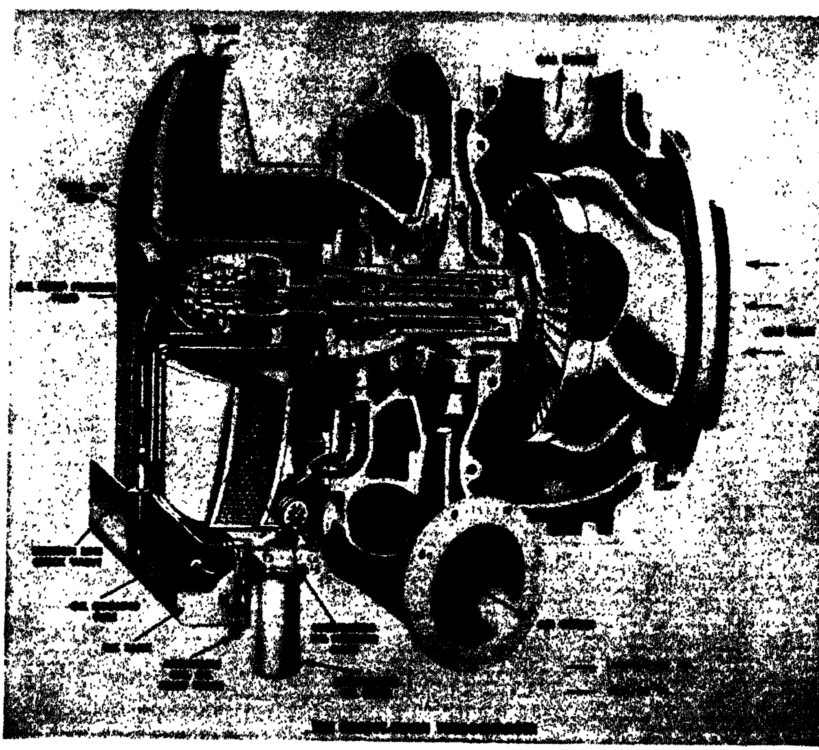
Details of Construction

Construction of the Elliott-Buchi turbocharger is shown in the cross-sectional view in Figure 3. Essentially, the turbocharger contains a rotor that is independent of the engine rotating elements, carries a single-stage impulse turbine driven by exhaust gases of the engine, and has a directly coupled single-stage centrifugal blower.

The engine exhaust gases are conducted to the cast-steel turbine inlet (159–4) by several exhaust headers. The number and arrangement of these headers are dependent on the number of engine cylinders. The turbine nozzle ring (107–4) is attached to the turbine inlet casing. The nozzle ring is cast of a special heat-resisting iron, in which are cast the alloy-steel nozzle blades.

The turbine casing (1–4) is a special iron casting, and is cored to provide cooling water passages. An oval flange is provided for the turbine exhaust-gas connection. Pads are provided at three points for supporting bracket connections.

The turbine casing backplate (287) is a special iron casting and is also cored to provide cooling water passages. This casting forms the wall separating the blower casing (163–4)



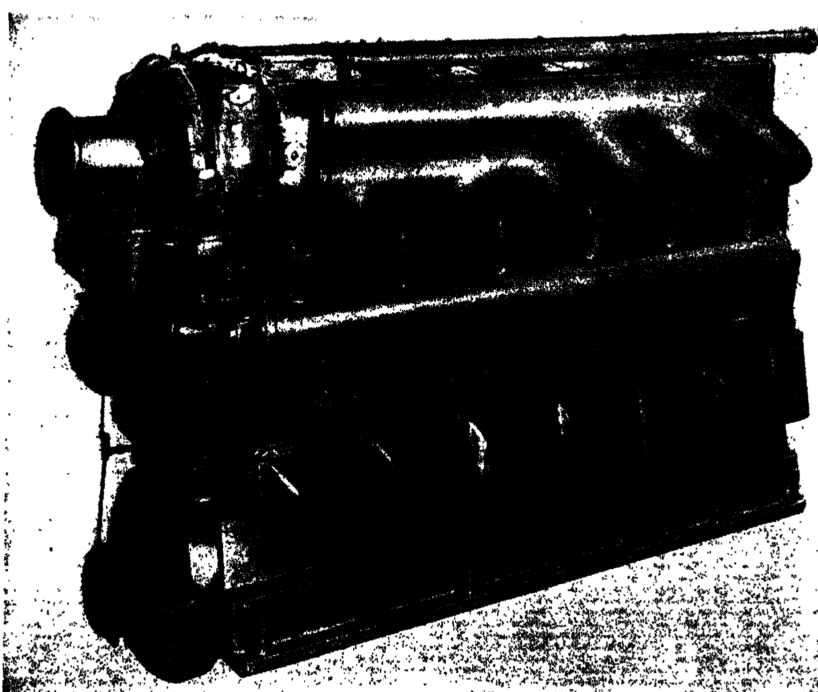
Courtesy of Elliott Company

Fig. 1. The Elliott-Buchi Turbocharger.

and turbine casing (1–4), and is split vertically to facilitate assembly. The backplate is attached on one side to the turbine casing by a ring of studs, and is attached on the other side to the blower casing by a ring of cap screws.

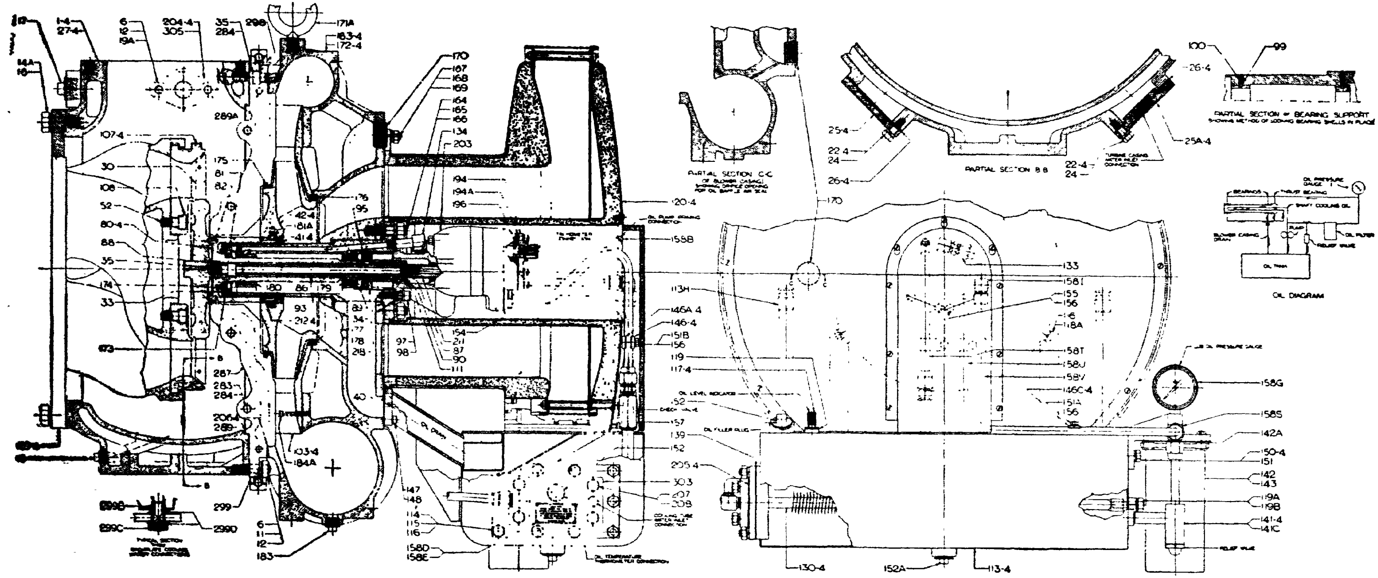
The blower casing (163–4) is a heavily ribbed iron casting. Air enters the blower casing axially and is discharged radially from the impeller through the diffuser ring (103–4) into the volute, which has a tangential discharge. The diffuser ring is a disk casting with special vanes to control passage of air. The diffuser ring is held in place by compression between the blower casing and the turbine casing backplate.

The shaft (80–4) is counterbored at the inner end to center to receive a projection on the turbine disk. The shaft is driven by two fitted dowels (82) on the turbine disk. The shaft is



Courtesy of Elliott Company

Fig. 2. Worthington Six-cylinder, 750-hp Diesel, Equipped with Elliott-Buchi Turbocharger. This is a typical installation.



Courtesy of Elliott Company

1-4—Turbine Casing
6—Lockwasher—Backplate to Turbine Casing
11—Stud—Backplate to Turbine Casing
12—Stud Nut—Backplate to Turbine Casing
14A—Hex. Hd. Mach. Bolt—Inlet Casing to Turbine Casing
16—Lockwasher—Inlet Casing to Turbine Casing
17—Pipe Plug
18—Pipe Plug
19A—Stud—Turbine Casing Water Discharge
22-4—Hex. Hd. Mach. Bolt—Cover Plate
24—Lockwasher—Cover Plate
25-4—Cover Plate
25A-4—Water Inlet Cover
26-4—Gasket
27-4—Turbine Casing Assembly Pcs. Nos. 1-4, 6, 11, 12, 14A, 16, 17, 18, 19A, 22-4, 24, 25-4, 25A-4, 26-4
30—Disk
33—Oil Baffle Nut
34—Thrust Bearing—Includes Piece 96
35—Socket Hd. Cap Screws—Backplate to Blower Casing
41-4—Impeller Nut
42-4—Impeller
52—Socket Hd. Cap-Screw—Nozzle Ring
54—Hex. Key for ½" Cap-Screw (Long Arm)
55—Hex. Key for ½" Cap-Screw
56A—Hex. Key for No. 10 Cap-Screw
57—Hex. Key for ¼" Set-Screw
59—Hex. Key for ½" Set-Screw
60—Bearing Shield
61—Rotor Blocking Rig
63—Wrench for Oil Baffle Nut
64—Hex. Key for ⅜" Cap-Screw
65—Wrench for Oil Pump Pipe Connection

65D—Hex. Key for ⅝" Cap-Screw
65G—Bearing Inserting Tool
80-4—Shaft
81—Slinger
82—Dowel—Disk—Shaft
86—Locking Tube
87—Oil Pump Coupling
88—Washer—Disk—Shaft Cap-Screw
89—Thrust Collar
90—Special Nut
91—Pin—Bearing Support
92—Plug—Bearing Support
93—Bearing Support
94—Bearing Support Assembly—Pcs. Nos. 33, 91, 92, 93, 99, 100, 173, 174, 179, 180
95—No. 1 Woodruff Key—Thrust Collar
96—Key—Thrust Bearing
97—Socket Hd. Cap-Screw—Oil Pump to Bearing Support
98—Lockwasher—Oil Pump to Bearing Support
99—Hollow Set-Screw, Dog Point—Bearing Support
100—Hollow Set-Screw, Cup Point—Bearing Support
101—Jack-Screw—Bearing Support
103-4—Diffuser Ring
107-4—Nozzle Ring
108—Locking Pin—Nozzle Ring Cap-Screw
111—Locking Washer
113-4—Oil Tank Assembly—Includes Pcs. 113H, 114, 115, 116, 117-4, 118A, 119, 119A, 119B, 152, 152A.
113H—Vent Pipe Cover
114—Stud—Oil Tank to Cooling Tube Flange
115—Stud Nut—Oil Tank to Cooling Tube Flange
116—Lockwasher—Oil Tank to Cooling Tube Flange
117-4—Oil Level Indicator
118A—Hex. Hd. Mach. Bolt—Oil Tank to Silencer
119—Washer—Oil Level Indicator
119A—Washer—Special Mach. Bolt
119B—Special Mach. Bolt
120A—Silencer
130-4—Cooling Tube Assembly
133—Tube Connector—Oil Piping
134—Special Connector—Oil Piping
139—Gasket—Cooling Tube to Oil Tank
140—Gasket—Blower Casing Oil Drain
141-4—Relief Valve
141C—Relief Valve Spring
142—Oil Filter
142A—Oil Filter Gasket
143—Oil Filter Core
146-4—Clamp (Half)—Oil Piping
146A-4—Clamp (Half)—Oil Piping
146C-4—Clamp—Oil Piping
147—Socket Hd. Cap-Screw—Oil Drain
148—Lockwasher—Oil Drain
150-4—Hex. Hd. Mach. Bolt—Oil Filter to Oil Tank
151—Lockwasher—Oil Filter to Oil Tank
151A—Hex. Hd. Mach. Bolt—Oil Piping Clamp
151B—Hex. Hd. Mach. Bolt—Oil Piping Clamp
152—Pipe Plug
152A—Magnetic Drain Plug—Oil Tank
154—Gasket—Oil Pump Suction and Discharge
155—Hex. Hd. Mach. Bolt—Oil Pump Suction and Discharge
156—Lockwasher—Oil Pump Suction and Discharge
157—Check Valve
158B—Pipe Plug—Oil Pump Priming Connection
158D—Caution Plate
158E—Drive Screws—Caution Plate
158G—Lub. Oil Pressure Gauge
158I—90° Tube Elbow—Oil Piping
158S—Run Tee, Male—Oil Piping
158T—Oil Piping Assembly—Suction Line
158U—Oil Piping Assembly—Pump to Relief Valve
158V—Oil Piping Assembly—Oil Filter to Special Connector
159-4—Inlet Casing
160—Jack-Screw—Inlet Casing
163-4—Blower Casing
164—Stud—Bearing Support to Blower Casing
165—Stud Nut—Bearing Support to Blower Casing
166—Lockwasher—Bearing Support to Blower Casing
167—Stud—Silencer to Blower Casing
168—Stud Nut—Silencer to Blower Casing
169—Lockwasher—Silencer to Blower Casing
170—Pipe Plug—Blower Casing
171A—Eye Bolt—Blower Casing
172-4—Blower Casing Assembly—Pcs. Nos. 163-4, 164, 165, 166, 167, 168, 169, 170, 171A, 183
173—Gasket—Oil Baffle
174—Oil Baffle
175—Labyrinth Ring, Inner
176—Labyrinth Ring, Outer
177—Gasket, Inner—Bearing Support
178—Gasket, Outer—Bearing Support
179—Bearing Shell, Outer
180—Bearing Shell, Inner
181A—Hollow Set-Screw—Impeller Nut
182-4—Rotor Assembly—Pcs. Nos. 30, 35, 41-4, 42-4, 80-4, 81, 82, 86, 88, 89, 90, 95, 111, 175, 181A, 212-4
183—Pipe Plug—Blower Casing Drain
184-A—Socket Hd. Cap-Screw—Diffuser Ring
194—Tachometer Adapter—Oil Pump
194A—Gasket—Tachometer Adapter
196—Oil Seal—Oil Pump
203—Oil Pump Assembly
204-4—Gasket—Turbine Casing Water Discharge
205-4—Gasket—Cooling Tube Water Inlet and Outlet
206-4—Gasket—Blower Casing to Backplate
207—Hex. Hd. Mach. Bolt—Cooling Tube Water Inlet and Outlet Flange
208—Lockwasher—Cooling Tube Water Inlet and Outlet Flange
209A-4—Instruction Plate on Silencer
211—Retainer Ring
212-4—Key Impeller
213—Name Plate—Engine
214—Name Plate—Turbocharger
215—Name Plate Drive Screws
218—Gasket—Oil Pump Flange
280—Backplate, Right Half } Not Sold Separately
281—Backplate, Left Half } Not Sold Separately
282—Eyebolt—Backplate
283—Socket Hd. Cap-Screw—Backplate
284—Lockwasher—For Pcs. Nos. 35 and 283
287—Backplate Assembly—Pcs. 280, 281, 283, 284, 289A
288—Jack-Screw—Backplate
289—Gasket, Asbestos Cord—Turbine Casing to Backplate
289A—Dowel—Backplate
298—Cooling Water Piping Assembly—Pcs. Nos. 299B, 299C and 299D Are *Not* Included
299—Cooling Water Piping Assembly—Pcs. Nos. 299B, 299C and 299D Are *Not* Included
299B—Nipple—Cooling Water Piping
299C—Cap—Cooling Water Piping
299D—Gasket—Cooling Water Piping
303—Flange—Cooling Tube Water Inlet and Outlet
305—Companion Flange—Turbine Casing Water Discharge

Fig. 3. Sectional Assembly of Elliott-Buchi Turbocharger.

secured to the turbine disk by a socket-head cap screw (35) which is retained by the locking tube (86). This tube provides an inward passage for the shaft cooling oil. A section of the shaft, adjacent to the turbine disk, overhangs the bearing support and carries the blower impeller (42–4). The shaft drives the impeller through four keys (212–4). The impeller is secured on the shaft by the impeller lock nut (41–4) which is secured by a screwed dowel (181-A).

The turbine-disk assembly (30) consists of a slotted disk and inserted blades. The blades have a bulbous root and are pressed into the rotor disk and peened. The disk and the blades are constructed of a special heat-resisting alloy steel.

The blower impeller (42–4) is of the single-inlet, enclosed type. The impeller is a one-piece casting of a special aluminum alloy, having a serrated steel insert cast in the bore. A steel bushing is pressed into the impeller bore on the one side of the serrated insert. The serrated insert and the bushing are bored for a close fit on the shaft, and are keywayed for the four driving keys. Steel labyrinth rings are retained in the backplate and blower casings, and seal the impeller discharge.

After finish-machining, the impeller is balanced statically and dynamically. The balanced impeller is assembled with the turbine disk and shaft. This complete assembly is then balanced both statically and dynamically.

The bearing support (93) is carried in the blower casing and is supported and aligned by a press fit and an external flange. The bearing support fits closely in the blower casing, and is sealed on inner and outer fits by oil-proof synthetic-rubber rings (177 and 178). Inner and outer bearing shells (180 and 179) are pressed in the bearing support and secured by locking screws (99 and 100). The bronze oil baffle (174) is inserted in the inner end of the bearing support, is sealed by a synthetic-rubber gasket (173), and is secured by the oil baffle nut (33).

The bearing support is drilled axially for lubricating oil. Supply connection is made at the bearing support flange face. Bearing sealing air is led from the blower casing volute,

through a throttling orifice, and into one of the horizontal cored ribs. Air enters the bearing support through radial drilling that aligns with the horizontal cored rib, and passes along axial drilling to the oil baffle. Bearing sealing air is introduced into the annular space in the oil baffle, which lies around the shaft, inward from the oil slinger to the seal and scavenge oil in the bearing support.

The unbalanced part of the rotor end thrust is taken by the thrust collar (89) at the outer end of the shaft. The steel thrust collar has hardened thrust faces, is keyed to the shaft, and is secured by the special nut. The thrust bearing (34) is keyed in the bearing support and has a specially grooved surface to support high bearing loads. The end clearance, or axial float, of the rotor is determined by the amount of clearance of the thrust collar between the outer-bearing shell flange and the thrust bearing.

The air-intake silencer (120–4) is fabricated of sheet steel. The silencer mounts directly on the blower casing flange. The air passage walls are formed by the perforated steel plate, back of which is packed felt. Access to the oil pump and lubricating-oil piping is possible through a removable cover on the silencer face.

Each turbocharger is equipped with a completely self-contained lubrication system. The oil tank (113–4) is fabricated of sheet steel and is supported by the silencer and by the blower-casing drain fitting. A finned oil cooling tube (130–4) is immersed in the sump tank, and is supported by an exterior flange at one end and by a pin (119-B), at the inner end. The oil pump assembly (203) contains a reduction-gear train having a ratio of 9.596 to 1 for the lubricating-oil pump and tachometer drive, and is supported by the bearing support flange. The pump assembly is driven by the turbocharger shaft through an oil-resistant rubber coupling (87) encircled by a retainer ring (211). Lubricating-oil tubing is of steel. The ball-type check valve (157) and the relief valve (141-4) are mounted in the oil piping. The lubricating-oil filter (142) is of the replaceable, yarn cartridge type.

Installation

Initial Installation on Engine

In order to lift the turbocharger, a hook, which will properly support and balance the unit, should be passed through the eyebolt in the top of the blower casing. Under no condition should a rope or chain be used through the ribs in the blower casing air inlet because this may damage the rotor.

Surfaces coated with a rust-preventative compound should be cleaned with a solvent. Before mounting on the engine, gaskets of material suitable for high temperatures should be available for the flanges between the exhaust manifolds and the inlet casing. A gasket should be available for the blower casing discharge flange. Graphite and oil or some other suitable compound should be applied to bolt and nut threads that are subjected to the temperatures encountered in the exhaust lines.

That the turbocharger rotor turn freely is essential. To learn how easily the rotor turns, it should be rotated in a counterclockwise direction by reaching the hand into the turbine casing discharge connection. After mounting the unit on its brackets and after coupling turbine inlet connection and air lines, a final check through the turbine casing discharge connection should be made to ascertain whether the shaft is free.

The lower turbine casing flange and the bolting lugs on either side of the turbine casing should be the points of support of the unit. Mounting brackets of suitable strength and rigidity are to be supplied by the engine builder. The piping to the inlet, turbine, or blower casing, should not be depended upon for support of the turbocharger. Conversely, no appreciable weight of such piping can be supported by the unit. The engine builder will provide either expansion pieces or joints in the exhaust gas or air lines and to and from the turbocharger in order to prevent transmission of piping strains to the turbocharger.

The name plate, which lists the numbers of patents applicable to the turbocharger and other pertinent data, is directly attached to the turbocharger. Also supplied is a separate

name plate which lists the numbers of patents referring to application of the Buchi system of turbocharging to the Diesel engine. The second name plate should be attached to the engine in some readily visible location, preferably near the engine name plate.

Service Installation

Expansion joints or flexible piping should be installed in the turbocharger exhaust piping to relieve the unit of piping strains. Such flexible provisions must be properly supported to prevent excessive weight being carried by the turbocharger casing.

The installation of the exhaust piping should be made in a manner which precludes the turbine back pressure from exceeding the maximum back pressure allowed for that particular engine.

Water inlet connections should be made at two places: at the cooling-tube companion flange and at the bottom of the turbine casing. The preferred location of the turbine casing inlet is 180 deg from the water discharge adapter. No throttling device, such as a valve, should be installed between the two inlet water connections which will cause an unequal pressure at the two water inlets. The cooling system is designed to give the most efficient cooling when the water inlet pressures at the two water inlets are equal.

The water outlet connection should be made at the discharge adapter. A sight flow indicator should be installed in the outlet line to give positive indication of water flow through the unit. The water discharge line should be inclined upward to the main water discharge header or surge tank to vent the jackets and to allow cooling by convection after the unit is shut down. If feasible, a separate pressure source should be provided for this purpose. If the turbocharger water discharge line cannot be carried upward to the main header or surge tank, a vent valve or tube should be provided in the high point of the water discharge piping to prevent vapor lock or siphoning.

In order to drain the interior of the turbine casing of any

condensate, leakage, or water that is taken in through the exhaust lines, a nipple and valve should be fitted to the turbine casing drain tapped hole located in the boss on the lower side of the turbine casing. This tapped hole is designated on the turbocharger outline drawing.

In order to avoid cracking the water jackets and developing leaks in the water passages during freezing weather, a nipple and valve should be fitted to the turbine casing water jacket drain tapped hole in the boss on the lower side of the turbine casing, and provision should be made to drain the turbine casing backplate oil cooler system. Drainage of this cooler system can be accomplished by installing a tee and valve in the water inlet line adjacent to the companion flange, or by grading this inlet line downward to some drainage point in the system.

Cooling Water

The cooling water system performs three functions. First, it prevents excessive heat conduction from the turbine end of the unit to the blower end. Second, it removes the heat from the oil that is generated in the bearings. Third, it prevents distortion of the mounting surfaces of the turbine casing.

Circulation of the water is divided into two parallel systems which are to have common supply and discharge connections. One system consists of the oil cooler, a connecting pipe, and the backplate. The other system consists of the turbine casing alone. Direction of flow through the first system is through the cooling tube, into the bottom of the turbine casing backplate, out the top of the turbine casing backplate, and into the discharge adapter on the turbine casing. Direction of flow through the turbine casing is in the bottom and out the top into the discharge adapter, where the flows of the two systems join.

Water circulation through the turbocharger should be provided and regulated at such a rate that the temperature rise of the cooling water does not exceed 25° F at full engine load. This precaution will minimize the possibility of distortion of parts caused by unequal temperatures. The discharge temperatures should not exceed 180° F for clean soft water. However,

lower temperatures may be required by characteristics of the water used.

Installation of piping should be made in accordance with instructions in order to assure proper water circulation and to permit draining. In freezing weather, when the unit is not operating, the jackets and oil cooler should be completely drained to prevent damage.

Connections or plugs should be removed annually, the cooling-water jacket spaces inspected for scale or sludge, and any accumulation removed. If the cooling water used is not clean and soft, more frequent attention will be necessary.

Owing to the amount of heat in internal parts and walls of the turbocharger, it is recommended that the cooling water system be so designed that the turbocharger water jackets will be adequately vented. Cooling water circulation through these jackets after shutdown of the unit will be provided by convection until the unit has cooled, or by some separate pressure source. Otherwise, heat remaining in the unit may be sufficient to boil the water in the turbocharger jackets or in a small, closed system.

Lubrication System

The oil-flow diagram and general piping layout in Figure 1 indicate the general details of the self-contained lubrication system. The lubricating-oil pump is direct driven from the main rotor shaft, through a reduction gear train. Gears and bearings in the pump are oiled through drilled passages in the gears and in the pump shafts. The pump drive shaft is drilled for passage of cooling oil to the rotor locking tube.

The pump takes suction from the oil tank through a suction-line check valve, and for lubricating the journal bearings and thrust bearing discharges through the oil filter into the bearing support. Pressure of oil fed to the bearings is controlled by a relief valve in the pump discharge line, which by-passes excess oil back to the oil tank before passing through the filter.

Lubricating-oil passages in the bearing support, and the cored drain space in the lower vertical blower casing rib, are

shown in Figure 1. Drilling in the bearing support header feeds the thrust bearing and the two journal bearings.

Cooling oil is conducted to the inner end of the shaft through the locking tube, and returns to the drain along the space between the locking tube and the shaft bore. Cooling oil leaves the shaft in the thrust bearing drain space, and drops into the drain rib. The thrust collar key controls the alignment of drain holes in the shaft and thrust collar.

Lubricating oil is returned from the oil baffle and from inner and outer bearing bushings through cored drain passages in the bearing support. The bearing sealing air is admitted into an annular space in the oil baffle, flows over the oil slinger, and aids in sealing and scavenging oil in the bearing support. Lubricating-oil and cooling oil drains combine in the bearing support and are led into the cored drain rib. Sealing air is vented from the oil tank by risers. The drain pipe is flanged to attach to the blower casing flange, and conducts oil from the drain rib to the oil tank.

Lubricating-oil pressure should be set by adjusting the pressure relief valve to a pressure of about 15 psi at a turbocharger speed corresponding to full engine speed and load. This pressure will vary somewhat as the turbocharger speed varies.

Lubricating-oil level should be checked with the bayonet-type oil gauge on the tank and the oil level properly maintained. If the oil level is maintained too high, excessive frothing and dissipation of lubricating oil will result.

Lubricating oil should be added to the tank through the filler opening. Only new, clean oil, free from acid, dirt, water, and foaming tendencies should be used in the turbocharger. The oil used should be of high quality and should be of medium weight, similar to an SAE 30 grade, having a viscosity range of 185 to 225 SSU at 130° F. The desired oil temperature in the tank is 100 to 125° F, and should not exceed 160° F.

The need for an uninterrupted supply of lubricating oil to the turbocharger bearings and the use of a high quality of oil that is free from contamination cannot be stressed too emphatically. The high rotative speed of the unit and the extreme

heat to which it is subjected make such precautions essential for obtaining satisfactory operation.

Starting Instructions

1. Cooling water connections as described should be checked to assure proper cooling water circulation and to ascertain whether the proper valves are open. The turbocharger water jacket must be filled with water before the engine is started. If a separate source is available, water circulation should be started before operating the unit.

2. The lubricating-oil sump tank should be filled to the middle of the gauge range with lubricating oil that has a viscosity of 185 to 255 SSU at 130° F, or an SAE 30 designation.

3. The lubricating-oil filter shell should be dropped and filled with lubricating oil.

4. The lubricating-oil pump must be primed each time the lubricating-oil lines have been broken or when the unit has been standing for an appreciable time. Access to the ⅛-in. priming plug (158B), which will be found in the pump suction line connected to the left side of the pump, is obtained by removing the upper half of the cover found on the front of the silencer. In priming, direct the oil toward the pump and fill the line to the check valve. The ⅛-in. plug must be replaced tightly because a leak at this point may cause the pump to lose its prime.

5. Start the unit and operate idle or at low speeds and loads (turbocharger should be at 2000 to 4000 rpm). Check the oil pressure gauge to see that the oil pressure develops properly.

6. If lubricating-oil pressure does not develop in 20 sec, shut the unit down and determine the cause of oil pressure failure.

7. Lubricating-oil pressure is set at 15 psi at full-load operating speed by relief-valve adjustment during the factory test of each turbocharger. This pressure will decrease normally at reduced turbocharger speeds, and should be five to seven pounds per sq in. at 2000 to 4000 rpm. Lubricating-oil pressure is adjusted by removing the relief-valve acorn nut, loosening

the lock nut, and running up the slotted adjusting screw to raise the pressure, or backing down the screw to lower the pressure.

8. Check for rubs by placing the end of a screw driver on the unit and holding your ear against the handle of the screw driver. If a rub develops, shut down the unit and check for piping strains.

9. Check the lubricating-oil pressure and temperature, and cooling-water jacket pressures and temperatures, shortly after the unit has been started, for proper flow conditions.

10. The turbocharger is ready for continued operation if the above starting precautions have been observed.

11. Operating conditions must not exceed those specified on the turbocharger name plate. Excessive exhaust-gas temperature before the turbine may be caused by:

- (a) Engine conditions:
  - (1) Stuck piston rings
  - (2) Spray-nozzle difficulties
  - (3) Valve conditions
  - (4) Timing of valves or fuel injection
  - (5) Low compression pressure
  - (6) Air-manifold leaks
  - (7) Exhaust-manifold leaks
  - (8) Poor combustion
- (b) Turbocharger conditions:
  - (1) Low discharge pressure
  - (2) Clogged air intake
  - (3) Excessive back pressure on turbocharger exhaust
  - (4) Impeller and diffuser clogged with dirt
  - (5) High inlet air temperature

Excessive turbocharger speed may be caused by:

- (a) Engine conditions:
  - (1) Smoky exhaust
  - (2) Incorrect valve timing
  - (3) Air-passage restrictions

(b) Turbocharger conditions:
  (1) High discharge pressure due to restriction in blower discharge piping
  (2) Clogged air intake
  (3) Impeller and diffuser clogged with dirt

After the turbocharger has been run 10 hr, a rotor runout or stopping period of about 1½ min is to be expected when the unit and oil are warmed up and when the turbocharger has been operating at ¼ to ⅓ speed. This speed range will correspond to light-load conditions.

Service Operation

Engine log data taken hourly should be supplemented by sufficient information on the turbocharger to permit observation of performance of the unit, and particularly to detect any change in performance. If an hourly engine log is not kept, performance of the turbocharger should be observed at intervals of four hours of operation. Data to be recorded and conditions to be observed are as follows:

Oil Pressure

Lubricating-oil pressure should be set at 15 psi at the turbocharger speed corresponding to full engine load. A marked increase in this pressure indicates faulty action of the pressure relief valve. A decrease in oil pressure indicates a need for changing the lubricating-oil filter cartridge, stoppage in feed lines, or faulty action of the relief valve, although some variation in lubricating-oil pressure with variation in speed is to be expected.

Oil Temperature

Temperature of lubricating oil supplied to the blower should not exceed 160° F, and lubricating oil at the drain should not exceed 180° F. Increase in lubricating-oil temperature may be attributed to loss in pressure, stoppage of internal passages, or inadequate cooling water circulation.

Rotative Speed

The rotative speed of the turbocharger is sensitive to the exhaust-gas temperature and condition. A high exhaust temperature and smoky exhaust will increase the turbocharger speed. Excessive accumulations of dust and dirt on the impeller and diffuser may cause an increase in rotative speed. A decrease in delivery pressure, such as is caused by a leak in air-intake manifold connections, will decrease the blower speed. The rotative speed of the turbocharger will be slightly affected by atmospheric conditions. For instance, the rpm of the unit will be increased over normal by a high inlet air temperature, or a low barometric pressure.

The unit must not be operated above the maximum allowable rpm indicated on the name plate. If there is a tendency for the speed to exceed the specified limit, the engine load should be reduced or the engine should be shut down and the cause remedied.

Exhaust-gas Temperature

In addition to the engine exhaust elbow temperature recordings, the temperature of the exhaust gas in each of the cylinder exhaust manifolds, at a point just before the turbine inlet casing, should be logged. If this temperature exceeds the temperature specified as a maximum on the name plate, the engine load should be reduced or the engine should be shut down and the cause remedied.

Water Temperatures

Water temperatures at the turbocharger supply and discharge connections should be recorded to keep a check on proper cooling of the turbocharger. If the water system design permits, cooling water should be circulated through the turbocharger jackets for a short time after the unit has shut down, until the unit has cooled reasonably.

Vibration

Operation of the unit should be observed frequently to detect any increase in vibration. If an appreciable increase in

vibration should develop, the unit should be shut down, and the cause determined. Vibration might be caused by loosening of the thrust-collar retaining nut, damage to the impeller, shaft, or turbine disk, or by loose bearings in the turbocharger or oil pump.

Any specific deposit on the rotor might contribute to vibration. In addition to the operating comments given in the preceding list, the following steps should be taken:

Lubrication

The lubricating-oil tank and the screen at the suction-line foot should be cleaned and flushed thoroughly after the first 200 operating hours, and thereafter at intervals of 2000 operating hours. The lubricating oil should be renewed after each such cleaning of parts. The lubricating-oil filter cartridge should be renewed, not cleaned, every 2000 operating hours, or oftener if this is required to maintain proper lubricating-oil pressure. The oil level should be checked frequently, and new oil added as required to bring the level in the oil tank up to the level range indicated on the bayonet gauge in the oil tank. Lubricating-oil consumption exceeding one quart per 24 operating hours should be investigated.

Operation of the turbocharger lubrication system will otherwise be automatic, and no attention will be required unless changes in the feed pressure or operating temperatures indicate need for investigation.

Bearings

Inspection of the turbocharger journal and thrust bearings should be made after the unit has been in operation 100 hr, 1000 hr, and thereafter at intervals of 3000 hr. Disassembly and reassembly of the bearing support should be made as discussed. Clearances should be checked against those listed. Bearings should be replaced if surface condition shows severe pitting or scoring. Slight evidence of wiping of the thrust bearing is not objectionable provided that the clearances are within the limits specified.

Impeller and Diffuser Cleaning

Accumulations of dirt and dust will be noticeable on the impeller and diffuser surfaces, even in apparently clean atmospheres. Such accumulations will develop especially on the under side of the impeller vanes, and should not be allowed to reach $\frac{1}{16}$ in. in depth. The required frequency of cleaning will depend on the operation atmosphere, and will vary, but cleaning after each 2000 operating hours or less will probably be required. The impeller and diffuser should be cleaned when the blower air-discharger pressure drops off sufficiently to affect engine operation and increase the exhaust temperatures.

Complete Turbocharger Inspection

It is recommended that the turbocharger be completely inspected and cleaned at least once a year.

Turbine Casing Drainage

When the engine and turbocharger are in continuous operation, there should be no collection of water in the turbine casing interior. However, water may accumulate in this space during shutdown, owing to condensation, introduction of water through the exhaust lines, or leaky gaskets. Before starting up after any but a brief shutdown, the turbine casing drain should be opened and any water collection drained. If lubricating oil collects in the turbine casing, check for oil leakage between the turbine disk and the shaft.

Emergency Operation

Should an accident or failure of some part of the turbocharger prevent or render inadvisable operation of the unit, the engine can still be operated until such time as repairs can be made on the turbocharger. Such a condition might prevail if damage occurred to the impeller or turbine disk owing to contact with some solid object, sudden increase in vibration, failure of bearings, or failure of lubricating-oil supply.

If possible, the turbocharger rotor should be blocked to prevent further damage by using the rotor blocking rig, furnished with the tools, to prevent rotation of the impeller. Load on

the engine must be limited to prevent exhaust temperatures before the turbine from exceeding values noted on the name plate. If practicable, engine lubricating oil should be circulated through the turbocharger to cool internal parts. It will be necessary to continue circulation of cooling water through the back plate and turbine casing water jackets.

The rotor blocking rig is installed on one of the blower casing ribs after removal of the silencer and oil tank. Care must be exercised, when installing the rotor blocking rig, to make sure it is firmly clamped to the blower casing rib and that it is correctly engaged with the leading edge of the impeller blade.

Operation of the engine with the turbocharger unit blocked should be kept at a minimum. If the unit has been operated under such emergency conditions, complete disassembly and inspection must be made as specified before again placing it in operation.

TABLE 1

TYPE BF-26 TURBOCHARGER—CLEARANCES[a]

	Desired	Limit
Rotor axial movement or end float, with surface oiled	0.012 –0.014 in.	0.020 in.
Journal bearings:		
Inner shaft diam	1.307 –1.306 in.	
Inner bearing bushing ID	1.3095–1.310 in.	
Outer shaft diam	1.120 –1.119 in.	
Outer bearing bushing ID	1.1225–1.123 in.	
Clearance on diam, dry, inner	0.0025–0.004 in.	
Clearance on diam, dry, outer	0.0025–0.004 in.	
Labyrinth rings (175 and 176):		
Clearance on diam over impeller	0.038 –0.046 in.	
Radial clearance between turbine blade OD and nozzle ring, measured on blade tip angle cold	0.030 –0.050 in.	
Oil baffle bores:		
Threaded bore ID	1.842 –1.843 in.	1.850 in. max
Smallest bore ID	1.509 –1.511 in.	1.520 in. max

[a] Worn parts should be replaced, or adjustments made, to assure that operating clearances will not exceed above values.

TABLE 2

TYPE BF-34 TURBOCHARGER CLEARANCES[a]

	Desired	Limit
Rotor axial movement or end float, with surface oiled	0.012 –0.015 in.	0.025 in.
Journal bearings:		
Inner shaft diam	1.495 –1.494 in.	
Inner bearing bushing ID	1.4980–1.4985 in.	
Outer shaft diam	1.245 –1.244 in.	
Outer bearing bushing ID	1.2475–1.2480 in.	
Clearance on diam, dry, inner	0.003 –0.0045 in.	
Clearance on diam, dry, outer	0.0025–0.004 in.	
Labyrinth rings (175 and 176):		
Clearance on diam over impeller	0.038 –0.046 in.	
Radial clearance between turbine blade OD and nozzle ring, measured on blade tip angle cold with rotor toward nozzle ring	0.047 –0.060 in.	
Oil baffle bores:		
Threaded bore, ID	2.010 –2.011 in.	2.025 in. max
Bore projection, ID	1.759 –1.761 in.	1.770 in. max

[a] Worn parts should be replaced, or adjustments made, to assure that operating clearances will not exceed above values.

TABLE 3

TYPE BF-44 TURBOCHARGER—CLEARANCES[a]

	Desired	Limit
Rotor axial movement or end float, with surfaces oiled	0.014 –0.018 in.	0.025 in. max
Journal bearings:		
Shaft diam	1.870 –1.869 in.	
Bushing ID	1.874 –1.8745 in.	
Clearance on diam, dry	0.004 –0.0055 in.	
Labyrinth rings (175 and 176):		
Clearance on diam over impeller	0.038 –0.046 in.	
Clearance between end of turbine blade and nozzle ring with rotor shifted toward nozzle ring	0.054 –0.070 in.	
Oil baffle bores:		
Threaded bore ID	2.509 –2.511 in.	2.525 in. max
Smallest bore ID	2.196 –2.198 in.	2.210 in. max

[a] Worn parts should be replaced, or adjustments made, to assure that operating clearances will not exceed above values.

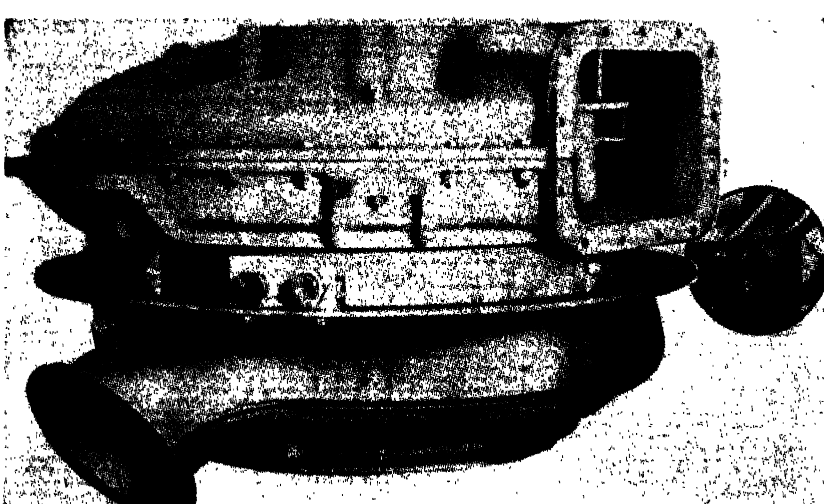
Chapter X

General Electric Company

The information following covers the description, operation, and service of the B-31 turbosupercharger (Fig. 1) designed and manufactured by the General Electric Company, Schenectady, N. Y.

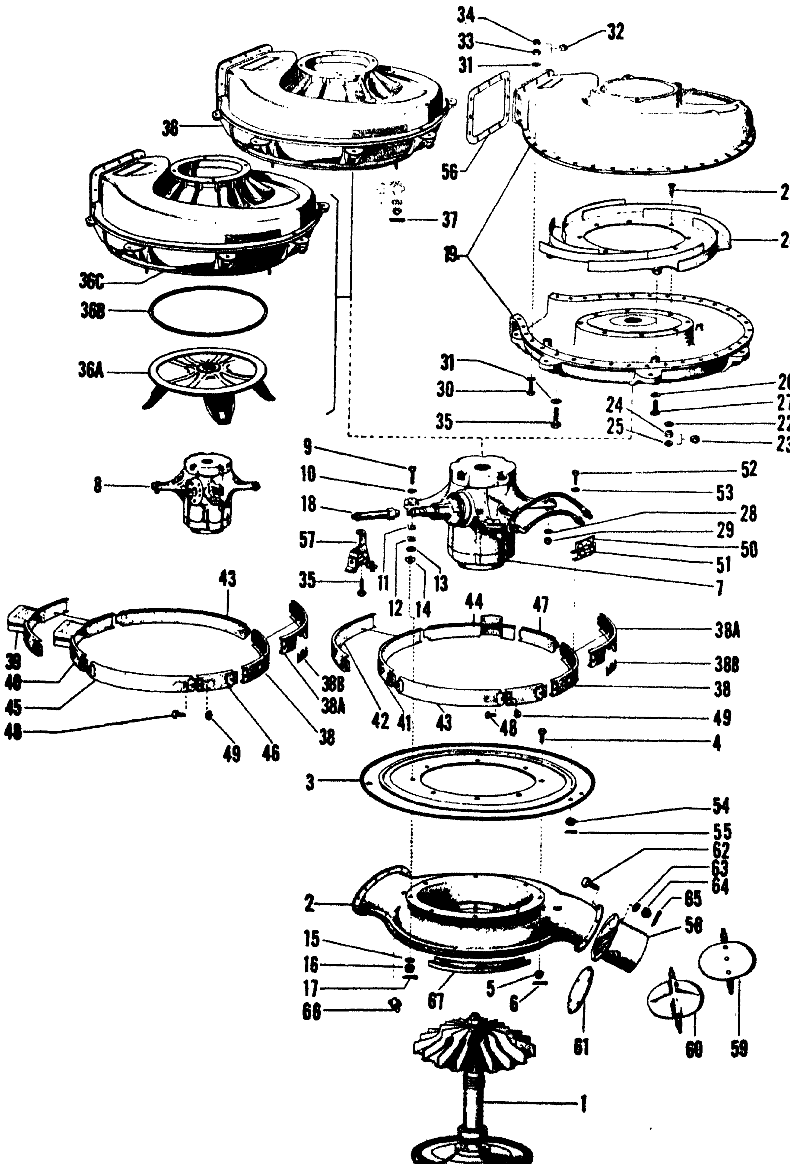
With slight differences peculiar to each type, this material may also be considered as adequately covering General Electric Types B-2, B-11, B-22, and B-33. The detailed description which follows uses the Type B-31 as a basic model. Model differences are pointed out where they exist.

The airplane-engine turbosupercharger is a variable-speed, centrifugal-type air compressor driven directly by a gas turbine, which is in turn driven by the energy of the exhaust gas



Courtesy of General Electric Company

Fig. 1. Assembly of Types B-11 and B-31 Turbosuperchargers.



Courtesy of General Electric Company

Fig. 2. Main Assemblies and Parts, Type B-31 Turbosupercharger.

List of Parts on Opposite Page

from the airplane's engine. The turbosupercharger makes possible high-altitude operation of the airplane engine because it compresses the thin atmosphere of the upper regions to approximately sea-level density for delivery to the carburetor, thereby maintaining normal intake-manifold pressure for the airplane engine.

KEY TO FIG. 2, PAGE 160

Number in parentheses after item indicates the number of units per assembly.

1—Rotor Assembly (1)
2—Nozzle Box Assembly (1)
3—Ring Assembly, Baffle (1)
4—Bolt (4)
5—Nut, Castle (4)
6—Pin, Cotter (4)
7—Casing Assembly, Bearing-and-Pump (1)
8—Casing Assembly, Bearing-and-Pump (1)
9—Bolt, Baffle Support (4)
10—Washer (4)
11—Shim (AR[1])
12—Shim (AR)
13—Shim (AR)
14—Washer, Lock (4)
15—Washer (AR)
16—Nut, Castle (4)
17—Pin, Cotter (4)
18—Drive, Tachometer Rigid (1)
19—Casing Assembly, Compressor (1)
20—Diffuser (1)
21—Screw, Flat Head (8)
22—Washer (8)
23—Nut, Self-locking (8)
24—Nut (8)
25—Nut, Lock (8)
26—Washer (4)
27—Bolt, Front-casing-and-diffuser (4)
28—Washer (4)
29—Nut, Castle (4)
30—Bolt (22)
31—Washer (58)
32—Nut, Self-locking (29)
33—Nut (29)
34—Nut, Lock (29)
35—Bolt (7)
36—Casing-and-cover-plate Assembly, Fabricated Compressor (1)
36A—Plate Assembly, Cover (1)
36B—Seal, Cover Plate (1)
36C—Casing Assembly, Fabricated Compressor (1)
37—Pin, Cotter (4)
38—Sector-and-plate Assembly, Lubrication Shroud (1)
38A—Sector Assembly, Lubrication Shroud (1)
38B—Plate, Lubrication-shroud Sector (1)
39—Sector Assembly, Shroud Inlet (1)
40—Sector Assembly, Shroud Inlet (1)
41—Sector Assembly, Tachometer Shroud (1)
42—Sector Assembly, Tachometer Shroud (1)
43—Sector Assembly, Plain Shroud (1)
44—Sector Assembly, Shroud Inlet (1)
45—Sector Assembly, Plain Shroud (1)
46—Sector Assembly, Plain Shroud (1)
47—Sector Assembly, Plain Shroud (1)
48—Bolt (5)
49—Nut, Self-locking (5)
50—Clamp, Oil Line (1)
51—Plate, Oil Line Locking (1)
52—Bolt (2)
53—Washer (2)
54—Nut, Castle (2)
55—Pin, Cotter (2)
56—Gasket, Compressor Discharge (1)
57—Bracket Assembly, Tachometer Drive Cable (1)
58—Wastepipe Assembly (1)
59—Wastegate-and-spindle Assembly (1)
60—Wastegate-and-spindle Assembly (1)
61—Plate Assembly, Blanking-off (1)
62—Bolt (6)
63—Washer (6)
64—Nut, Castle (6)
65—Pin, Cotter (6)
66—Nut-and-retainer Assembly (10)
67—Strip-assembly, Fabricated Nut (2)

[1] As required.

TABLE 1

TURBOSUPERCHARGER DESIGN CHARACTERISTICS

Type	Weight Flow Lb Per Min at Rated Altitude	Rotor Speed Rpm at Rated Altitude	Approx Wt, Lb
B-2[a]	110 at 25,000 ft	21,300	135
B-11[b]	120 at 28,000 ft	24,000	144
B-22[c]	120 at 28,000 ft	24,000	138
B-31[d]	120 at 28,000 ft	24,000	144
B-33[e]	130 at 27,500 ft	24,000	138

[a] This type is the basic turbosupercharger for a 1000-hp engine.

[b] Type B-11 is the same as the B-2 with the following exceptions: (a) ½-in. flexible-rubber oil lines, (b) coil spring in *oil in* lines, (c) antitwist clamps on oil lines, (d) redesigned bearing lubrication, (e) single-piece shroud support ring, (f) new-design nozzle-box sealing plate, (g) three notches cut in front compressor casing, (h) welded bucket wheel. and (i) two pairs of nut strips for mounting exhaust hood.

[c] Type B-22 is the same as the B-2 with the following exceptions: (a) bearing lubrication redesigned and bucket wheel of improved material permitting higher speed and (b) ½-in. flexible-rubber oil lines and antitwist clamps on oil lines.

[d] Type B-31 is the same as the B-11 with the following exceptions: (a) rigid drive shaft from oil pump projects outside cooling shroud (on setting No. 2 only), (b) reversed *L* ring on nozzle box, (c) support lugs under nozzle-box *L* ring, (d) bucket deflector ring on nozzle box, and (e) pair of nut strips for mounting exhaust hood.

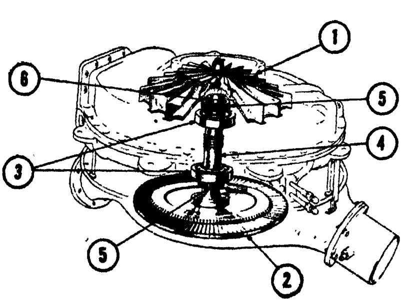
Note: Some late-type B-11 turbosuperchargers also have the nozzle-box construction covered by (b), (c), (d), and (e).

[e] Type B-33 turbosupercharger is the same as the B-22 except that the diffuser has been changed to satisfy the requirements of a 1100-hp engine and the wastepipe is cut at an angle.

Detailed Description

Main Assemblies

The three main assemblies of the Type B turbosupercharger are: the centrifugal air compressor, the bearing-and-pump casing, and the exhaust-gas turbine. These three main parts are assembled in various relative positions so that their openings will line up with the corresponding ducts and piping of the airplane in which the turbocharger is to be installed (Fig. 2).



Courtesy of General Electric Company

1—Impeller 4—Pump Drive Sleeve
2—Bucket Wheel 5—Oil Deflectors
3—Bearings 6—Impeller Spacer

Fig. 3. Rotor of Turbosupercharger.

Rotor

The heart of the turbosupercharger is the rotor unit (Fig. 3), with the impeller (1) of the centrifugal air compressor and the bucket wheel (2) of the exhaust-gas turbine mounted on opposite ends of the same shaft. A series of auxiliary parts, including two antifriction bearings (3), a pump-drive sleeve (4), two oil deflectors (5), and an impeller spacer (6), are mounted on the shaft between the impeller and bucket wheel to complete the rotor assembly.

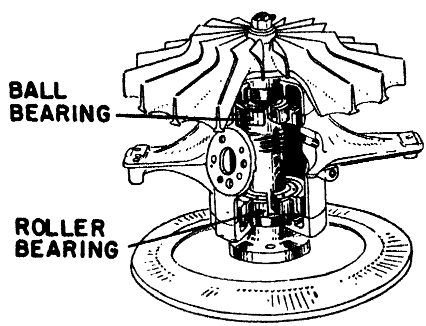


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Fig. 4. Location of Bearings in Casing.

Bearings

The rotor revolves on a ball bearing and a roller bearing, which are mounted on opposite ends of the bearing-and-pump casing (Fig. 4). The ball bearing supports one end of the rotor assembly and bears the entire thrust load and part of the radial load.

Lubrication System

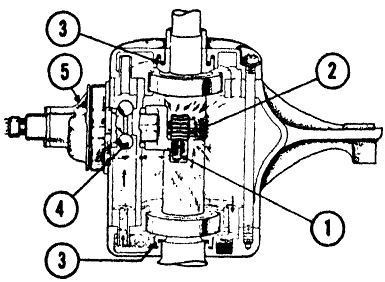
The built-in lubricating pump, which is a compact, high-speed, double-gear-type unit, supplies oil to the drive gears and rotor bearings (Fig. 5). It is situated in the bearing-and-pump casing of the turbosupercharger.

The lubricating pump is two separate, positive-displacement pumps on the same shaft. One element supplies oil to the gears and bearings, whereas the other element is a scavenging unit which removes oil from the casing and returns it to the supply tank.

Lubricating oil enters the inside of the bearing-and-pump casing through a shroud (1, Fig. 5). The oil is pumped through a jet to the mesh of the drive gear and the worm thread sleeve (2). Late-model turbosuperchargers are designed with jets which deliver oil directly on the ball and roller bearings. This results in no better lubrication than the oil mist.

but it does provide for more efficient cooling of the bearings and permits higher operating speeds.

The oil which collects in the bearing-and-pump casing is removed through a sump bushing by the larger gear pump, called the *scavenging pump,* the capacity of which is about three times that of the pressure pump. Because of this difference in capacities, approximately two thirds of the scavenging-pump delivery is air.



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1—Shroud
2—Drive-gear Assembly
3—Shaft Oil Seals
4—Dumbbell Valve
5—Pump

Fig. 5. Cutaway of Bearing-and-pump Casing.

The two shaft oil seals (3, Fig. 5), one on the turbine end and the other on the compressor end of the bearing casing, are not rubbing seals, but have a clearance from the shaft. These seals are threaded to cause an inward flow, which tends to keep the oil inside the bearing-and-pump casing. The result is an effective oil seal around the shaft.

The dumbbell valve (4, Fig. 5) operates by gravity. The intake of the scavenging pump is through this valve, and the valve position will always be such that the scavenging-pump intake will draw only from the bottom of the bearing-and-pump casing, regardless of the position of the airplane in flight.

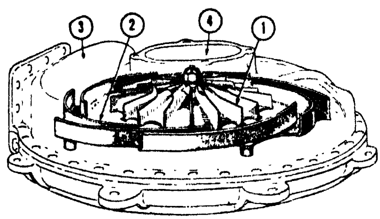
Compressor

The centrifugal air compressor is composed of an impeller (1, Fig. 6), a diffuser (2), and a compressor casing (3). Air enters through a circular opening (4) in the center of the rear compressor casing, is picked up by the impeller blades, and is given a high velocity as it travels outward toward the diffuser. The diffuser vanes straighten out the air flow and also serve to convert the velocity of the air into pressure.

Nozzle Box and Cooling Cap

Immediately adjacent to the compressor are the nozzle box of the exhaust-gas turbine and the cooling cap (Fig. 7). The

nozzle box end of the turbosupercharger is referred to as the *front* of the turbosupercharger. The hot exhaust gases enter the nozzle box and are directed onto the bucket-wheel blades (2) by means of a diaphragm (3) located in the nozzle box. Attached to the nozzle box is a cooling cap (4) which cools the



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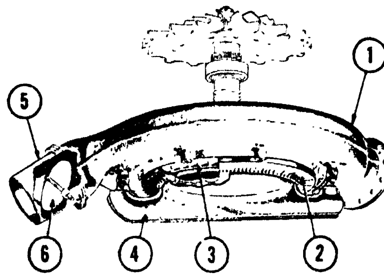
1—Impeller

2—Diffuser

3—Compressor Casing

4—Air Inlet

Fig. 6. Diffuser in Compressor Casing.



Courtesy of General Electric Company

1—Nozzle Box
2—Bucket-wheel Blades
3—Diaphragm
4—Cooling Cap
5—Wastepipe Assembly
6—Wastegate

Fig. 7. Gas Turbine Details.

bucket wheel. On all Type B turbosuperchargers, a waste-pipe assembly (5) containing the waste gate (6) is attached to the nozzle box.

Diaphragms. Diaphragms of Type B nozzle boxes are of two types, either fabricated (welded) or cast. Some fabricated nozzle boxes that have been overhauled have been modified by the addition of a partition, or dam, that reduces warping and cracking of the nozzle box. The one-piece cast diaphragm has been incorporated in late production models. Not requiring a dam, it is more efficient than the fabricated type, and is less susceptible to warping and cracking. The physical characteristics of the different styles of Type B nozzle boxes are as follows:

Nozzle Box with Fabricated (Welded) Diaphragm. This type of construction can be recognized by the beads of welds on the outside of the diaphragm ring which holds the individual nozzle blades in place.

Nozzle Box with Fabricated Diaphragm and Partition (Dam) in the Shell. This is the standard fabricated-diaphragm Type B nozzle box. A partition (dam) has been welded into place so as to prevent the hot gas from flowing directly into the narrowest section of the nozzle box. This partition can be seen by looking into the nozzle-box inlet opening and it can usually be distinguished, when the turbosupercharger is installed in an airplane, by looking for the welding line on the outside of the nozzle-box shell in the section where the partition is installed (Fig. 8).

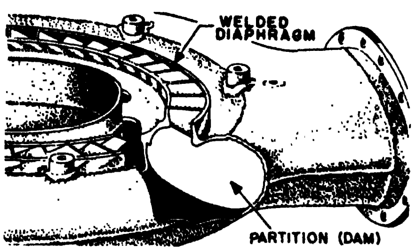


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Fig. 8. Welded Diaphragm and Dam in Nozzle Box.

Diaphragms and Operating Characteristics. These modifications of the nozzle boxes produce a slight difference in the turbosupercharger operating characteristics. Turbosuperchargers which are equipped with a nozzle box in which a partition has been installed will deliver a higher manifold pressure for a given waste-gate position than will those not so equipped. Also, because the cast diaphragm if used is more efficient in its operation than the fabricated diaphragm, turbosuperchargers equipped with nozzle boxes containing the cast diaphragm will deliver a higher manifold pressure for a particular boost-control lever setting than those equipped with nozzle boxes containing the fabricated diaphragm without the partition installed. Because of these differences in performance, it is desirable that all turbosuperchargers on multiengine installations have nozzle boxes of the same type.

Waste-pipe Restrictors. Certain turbosupercharger installations have waste-pipe restrictors, installed over the waste-pipe discharge (Fig. 9). The object of the restrictor is to obtain take-off manifold pressure at full throttle with the boost lever in the *off* position. The waste-pipe restrictor was conceived as a military convenience to eliminate loss of time at take-off, since a turbosupercharged airplane so equipped can take off

as quickly as a trainer with no danger of overboosting the engine. Where these are installed, it will be found that any part-throttle operation will be less efficient than an installation without the restrictors, although the differences for com-

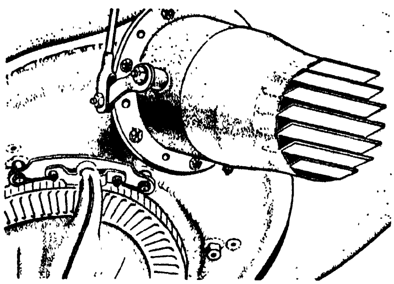

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Fig. 9—Restrictor installed on Wastepipe.

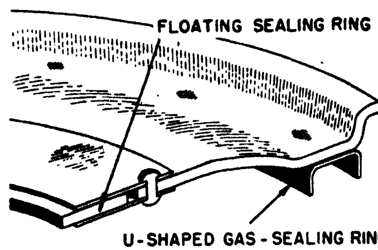

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Fig. 10. Modified Sealing-plate Assembly.

mon cruising operation will be negligible, and any full-throttle operation will be exactly the same as though no restrictors were present. Take-off manifold pressure can be controlled to compensate for the variation in altitude of different flying fields by bending the tabs on the restrictor.

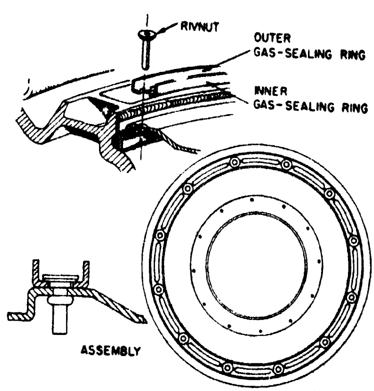

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Fig. 11. Split-type Gas-sealing Ring.

Sealing Plates. Types B-11 and B-31 turbosuperchargers have nozzle boxes with sealing-plate assemblies which are usually one of two designs:

The modified sealing plate has a single, floating sealing ring and a *U*-shaped gas-sealing ring welded to the plate (Fig. 10).

The second type of sealing plate incorporates the single, floating sealing ring (Fig. 10) and a split-type gas-sealing ring (Fig. 11). This type of sealing plate assembly is always used for replacement purposes. Turbosuperchargers that have this type of sealing plate installed can be identified by the fact that the legs of the waste-

pipe bracket have been ground off. This identification holds for all Type B-31 turbosuperchargers except AAF setting Nos. 5 and 6 for the B-32 airplane. In this case, the legs of the waste-pipe bracket extend slightly above the feet of the bracket which indicates that the sealing-plate assembly does not incorporate the split-type gas-sealing ring.

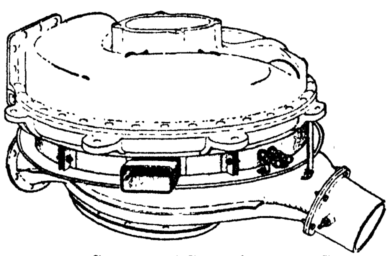

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Fig. 12. Cooling Shroud.

Cooling Shroud

Surrounding the area between the baffle ring and the compressor casing of the Types B-11 and B-31 turbosuperchargers is a cooling shroud (Fig. 12). Outside ram air is forced into this region to cool the bearing-and-pump casing.

Common Features

All Type B turbosuperchargers have the following features in common:

1. A single compressor air inlet and a single compressor air discharge.
2. A turbine driven from the engine exhaust gases.
3. A bucket wheel and an impeller both mounted on one shaft and supported on antifriction bearings.
4. A bucket wheel which is air-cooled by means of a cooling cap.
5. A lubrication pump which is gear-driven from the rotor shaft and which contains both the scavenging and pressure-pumping units of the circulating oil system.
6. Counterclockwise rotor rotation (facing bucket wheel).
7. Clockwise lubrication pump rotation (facing tachometer drive).

Installation

The information contained in this section is of a general nature. For details on the procedure for turbosupercharger installation in or removal from a particular airplane, reference

should be made to instructions covering the specific airplane involved.

The standard Type B turbosupercharger consists of three main assemblies: compressor casing, bearing-and-pump casing, nozzle box and, on the B-11 and B-31 models, a cooling air shroud. These assemblies are so bolted together that the relative positions of the various openings will line up with the corresponding ducts and piping of the airplane into which the unit is to be installed. A definite parts position, or relationship, of openings is required, and the Army Air Forces gives each parts position for each type of turbosupercharger an AAF setting number. For example, Type B-11 (or B-31) turbosuperchargers in the B-29 airplane are assembled to AAF setting Nos. 1 and 2, depending on the particular side of the nacelle involved, and some forms of these turbosuperchargers can be installed only in B-29 nacelles that incorporate a nacelle change that concerns repositioning of the turbosupercharger governor in the nacelle.

Service Inspection, Maintenance, and Lubrication

The work outlined in this section is what is required of the ground crews in the way of periodic inspections and maintenance to keep the turbosupercharged installation performing satisfactorily under the extreme conditions encountered during high-altitude flying.

Although it is relatively simple in principle and construction, the turbosupercharger operates at temperature, pressure, and speed ranges that impose severe strains on the unit. The entire turbosupercharger installation requires the same attention during maintenance and servicing that is called for on any precision equipment, and the following procedure will show why this care is necessary, when it should be done, and how the details of inspection and maintenance can be quickly and efficiently completed.

Service Inspection—Daily and Preflight

1. Check the rotor to see that it rotates freely.
2. Inspect the buckets for mechanical damage, mushroom-

ing, cracks, necking, stretching, back-lean, axial displacement, and gap. Check for broken buckets.

3. Inspect the wheel blank for cracks and other evidence of failure.

4. Inspect the nozzle box and cooling cap for loose bolts, broken safety wires, stretched bolts, and security of mounting.

5. On installations equipped with deflector rings mounted on the nozzle box, inspect the ring for cracks, warpage, and security of mounting.

6. Inspect the nozzle box for cracks and evidence of deterioration.

7. Check the clearance between the nozzle box and the bucket wheel (Fig. 13).

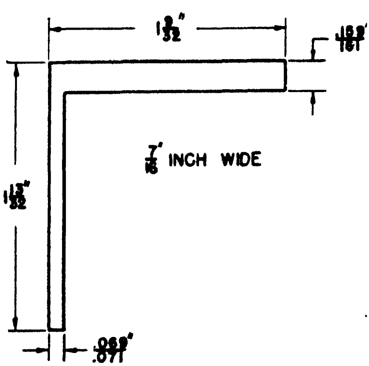

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Fig. 13. Nozzle Box Clearance Gage.

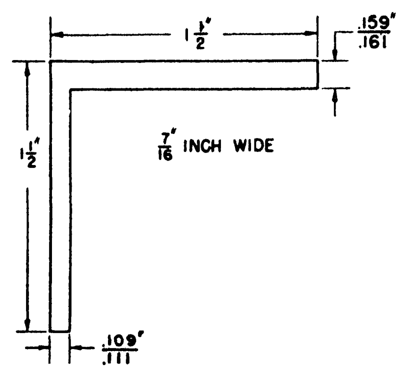

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Fig. 14. Cooling Cap Clearance Gage.

8. Check the clearance between the cooling cap and the bucket wheel (Fig. 14).

9. Inspect the waste gate for freedom of operation.

10. Check for excessive slop in the linkage system.

11. Check the oil supply.

12. Check the lubrication system for leaks and evidence of chafing, wear, or failure.

13. Inspect the exhaust-manifold system for cracks or leaks, and see that the system is properly supported.

14. Inspect for obstructions in the air scoop.

Twenty-five-hour Inspection

1. Examine the ducts, joints, and gaskets of the air induction system.
2. Check the exhaust-manifold bands and flexible connections for freedom and alignment.
3. Check the flexible drive shaft for security, and see that the nut on the shaft is not cross-threaded.

One-hundred-hour Inspection

1. Check the total side (radial) play of the rotor.
2. Check the total end (axial) play of the rotor.
3. Check the bucket-wheel runout.
4. Lubricate and inspect the flexible drive shaft.
5. Pressure test the induction system.
6. Inspect the turbosupercharger for insecurity or signs of damage or wear.
7. Remove, disassemble, and clean the oil filter.
8. Inspect the air filters and clean them, if necessary. When operating under extremely dusty conditions, inspect the air filters after each flight, cleaning them, if necessary.

Maintenance

Operating Time

The operating time between overhauls for turbosuperchargers should not exceed that specified in Table 2.

TABLE 2

Type of Turbosupercharger	Engine Series	Max Flying Hr
B-2	V-1710	750
B-2	R-1820 and R-1830	1300
B-11	R-3350	1300 [a]
B-22	R-1820 and R-1830	1300
B-31	R-3350	1300
B-33	V-1710	1300

[a] Permissible running time for B-11 turbosuperchargers on which the nozzle boxes have not been modified is 360 hr. Turbosuperchargers with modified nozzle boxes can be identified by a one-inch blue band around the compressor casing.

Freedom of Rotor

Internal rubbing or indication of bearing failure can be detected by spinning the bucket wheel counterclockwise by hand. Replace the turbosupercharger if any defects are noted.

Bucket-wheel Defects

Mechanical Damage. Nicks, dents, or gouges on the underside (rear) of the buckets (Fig. 15) are caused by foreign ma-

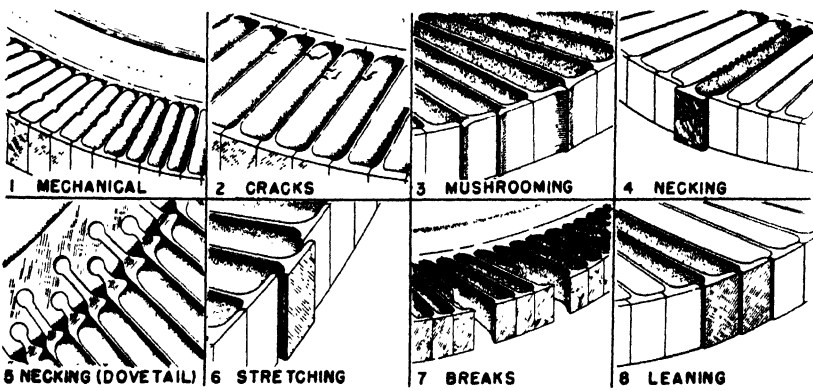

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Fig. 15. Bucket Wheel Defects.

terial coming in contact with the rotating bucket wheel. This usually signifies that loose nuts, bolts, or other stray objects have entered the exhaust system and have been discharged through the diaphragm onto the bucket wheel. Such damage is cause for removal of the turbosupercharger from the airplane.

Cracks. Cracked buckets are cause for removal of the turbosupercharger from the airplane.

Mushrooming or Upsetting. If the bucket wheel assembly is such that the total gap between the ends of the bucket shrouds is less than that required to absorb the expansion of the wheel periphery at operating temperature, the buckets will press together, thus causing an upturning, or *mushrooming,* of the ends of the bucket shrouds. Mushrooming is not a cause for removal of the turbosupercharger unless it has progressed to the point where the end of the shroud is cracked or broken.

Necking (Welded Wheels). In welded wheels, necking of the bucket occurs only in the blade. Necking is distinguished by a contraction in the cross section of the blade, and is cause for removal of the turbosupercharger.

Necking (Dovetailed Wheels). Necking of the bucket may occur in the blade (similar to (4), Fig. 15), or it may localize in the neck of the bucket (5). Check also to see if the wheel blank metal is distorted.

Stretching. Centrifugal force and high operating temperatures may cause elongation or stretching of the buckets. When this has progressed to the point where the shroud of one bucket extends beyond the shroud of adjacent buckets, the turbosupercharger must be replaced.

Breaks. Broken buckets affect the balance of the rotor beyond safe operating limits and are cause for removal of the turbosupercharger.

Back-lean or Tipping. Overspeeding at the high speeds and temperatures under which turbine wheels operate may cause a slight leaning, or tipping, of the buckets. This gives the circumference of the wheel a jagged appearance, but it is not cause for removal of the turbosupercharger unless it is so pronounced that the shroud of one bucket overlaps that of the following bucket.

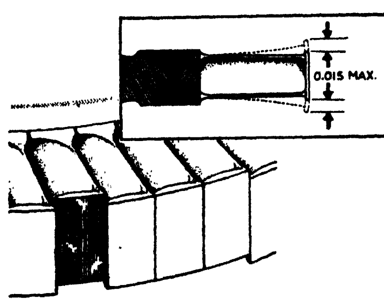

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Fig. 16. Axial Displacement.

Gap Between Adjacent Buckets. Buckets are assembled in the wheel disk with an accumulative nominal clearance between the ends of the bucket shroud to allow for expansion. This clearance on a factory assembly is equally spaced around the wheel, but owing to leaning and mushrooming, a series of buckets may be found with their shrouds touching and having a large gap between this series and the next. This condition is normal, and is to be disregarded unless other defects necessitate removal of the turbosupercharger.

Axial Displacement. Buckets may become displaced axially under high speed and temperature conditions. This will give the edge of the shroud band a jagged, or *sawtooth,* effect (Fig. 16). Axial displacement must not exceed a maximum of 0.015 in. from the normal in either direction. If this limit is exceeded, replace the turbosupercharger. This condition is more likely to be found on welded wheels than on other types.

Cracks in Welded Wheels

Radial cracks in welded wheels which extend into the wheel blank metal within a circle of approximately 9⅛ in. diam are cause for removal of the turbosupercharger. The distance between the outside of the shroud band and this imaginary circle is 1⅝ in. On late-serial turbochargers, there is a raised area in the wheel blank (Fig. 17) that corresponds to these limits. Radial cracks must not extend into this raised area. Some bucket wheels may have an etched circle on the wheel blank, and radial cracks must not extend past this etched circle. Cracks in welded wheels which extend in other than a radial direction must not progress more than 3/16 in. circumferentially even though they are outside the specified limits (Fig. 17).

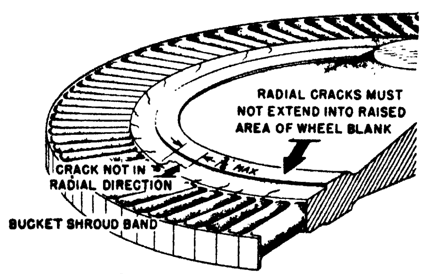

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Fig. 17. Radial Cracks in Bucket Wheel.

Bucket-wheel Deflector Ring

On installations equipped with a deflector ring mounted on the nozzle box, inspect the ring for cracks, warpage, and security of mounting (Fig. 18).

1. Visually inspect the eight mounting bolts for stretching and looseness. Stretched bolts must be replaced. Bolts should be tightened snug-tight at installation. Be sure that all clamps are tight and that there are no broken safety wires.

2. Heat may cause the deflector ring to warp and thus dis-

tort the nozzle-box diaphragm. Replace the deflector ring if the warpage has seriously distorted the nozzle box or diaphragm. Thoroughly inspect the weld around the base of each of the mounting bosses for excessive cracking.

3. Cracks or gouges will not be cause for replacement of the deflector ring unless the ring is split or ruptured.

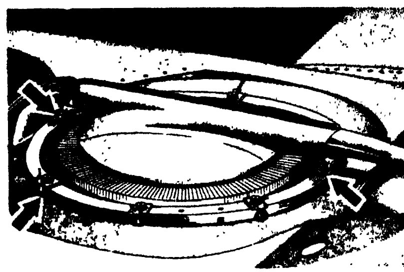

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Fig. 18. Inspection of Deflector Ring.

Visual Inspection of Nozzle Box for Cracks (Fig. 19)

Thoroughly inspect the entire nozzle box for cracks (especially in the weld) and for evidence of deterioration.

1. The inside of the nozzle diaphragm can be inspected by spinning the bucket wheel and projecting a flashlight beam into the clearance between the nozzle diaphragm and the bucket wheel and looking through the space between the buckets.

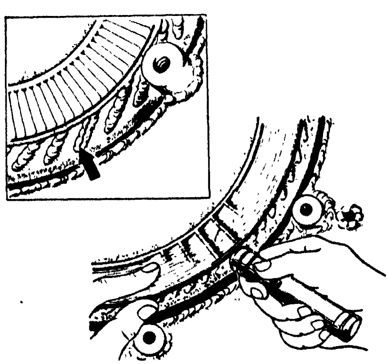

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Fig. 19. Inspecting Diaphragm for Cracks.

2. If cracks more than one inch long are observed in the blade weld of a fabricated diaphragm, or if any cracks are observed in the walls of the nozzle box which would result in leakage, replace the turbosupercharger.

3. On turbosuperchargers having fabricated diaphragms, see if there is any buckling of the diaphragm nozzle blades, especially toward the wheel. Also look through the clearance space between the buckets of the turbine wheel and nozzle diaphragm for possible buckling. If any buckling is observed, replace the turbosupercharger.

4. On *submerged* installations (as, for example on B-29

or B-32 airplanes) the inspection may be made during the 25-hr inspection unless failure of the nozzle box is suspected. When the inspection is made on this type of installation, sufficient fairing must be removed to expose the entire nozzle box.

Nozzle Box Clearance

1. Measure the maximum and minimum clearance between the bucket wheel and nozzle box at four different places by inserting a feeler gauge radially into the clearance far enough to measure the clearance of both the inner and outer rings of the nozzle box diaphragm (Fig. 20). If the maximum clearance at any point is greater than 0.160 in., or if the minimum clearance is less than 0.070 in., reshim the turbosupercharger. If the reshimming cannot be accomplished with the turbosupercharger installed in the airplane, remove the unit for reshimming at a repair station, and replace it with a satisfactory unit.

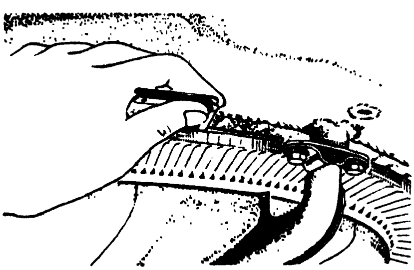

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Fig. 20. Checking Nozzle Box Clearance.

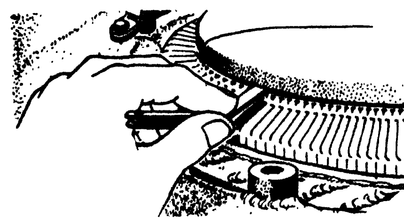
2. On submerged installations using the Type B-11 or B-31 turbosupercharger, this clearance check is made through the exhaust hood by using a special L-shaped nozzle box clearance gauge (Fig. 13). Replace the turbosupercharger if the clearance is outside the 0.070- to 0.160-in. range.

Cooling Cap Clearance

1. Measure with a feeler gauge the maximum and minimum clearance between the cooling cap and the turbine bucket wheel at four or more places (Fig. 21). If the maximum clearance is greater than 0.205 in. or the minimum clearance is less than 0.090 in., change the shims supporting the cooling cap until the clearance is within these limits around the entire circumference of the cooling cap rim.

2. On submerged installations using the Type B-11 or B-31

turbosupercharger, this clearance check is made through the exhaust hood by using the special clearance gauge (Fig. 14). The maximum clearance allowed for this type of installation, which uses the radiation-type cooling cap, is 0.0160 in., and the minimum clearance allowed is 0.110 in. If the clearance at any point between the cooling cap and the bucket wheel is not within these limits, the shims supporting the exhaust hood must be adjusted until the clearance is satisfactory. Carefully inspect the bucket wheel for any grooves or gouges caused by the cooling cap rubbing on the bucket wheel. Replace the turbosupercharger if any such grooves, no matter how slight, are found.



Courtesy of General Electric Company

Fig. 21. Checking Cooling Cap Clearance.

Waste Gate

Visually and manually inspect the turbosupercharger waste gate and linkage to see that it operates freely. The end play of the spindle should be approximately $\frac{1}{16}$ in. Inspect both the spindle and the gate to see that they are not distorted or warped because such defects may cause binding or unsatisfactory operation. If the spindle is bound with carbon, saturate the bearings with a carbon-removal compound and work out the carbon by moving the spindle from side to side.

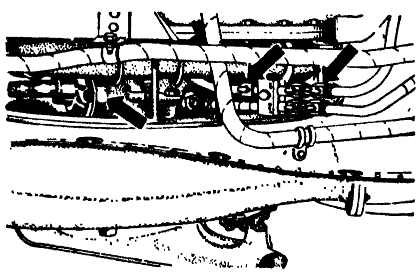
Oil Supply

Remove the cap from the turbosupercharger oil-supply tank and check the level of the oil in the tank. If the oil is not up to the bottom of the filler neck (three-quarters full), add oil until the tank is three-quarters full. This check is not necessary on airplanes where the lubricating oil for the turbosupercharger comes direct from the engine oil supply system.

Leaks in the Lubrication System

Using a flashlight, check as much of the turbosupercharger lubrication system as possible for leaks and for security of the

mounting (Fig. 22). If leaks are found or if there is any evidence of chafing, repair or replace the faulty parts. Oil seepage from the turbosupercharger oil seal onto the bucket wheel and cooling cap is permissible when the turbosupercharger is idle.



Courtesy of General Electric Company

Fig. 22. Possible Points of Oil Leakage.

Leaks in the Exhaust-manifold System

At sea level, where the pressure difference between the gases in the exhaust system and the atmosphere is slight, leaks do not seriously affect the performance of the power plant. However, this condition changes radically as the airplane gains altitude and the atmospheric pressure decreases. With the pressure of the hot gases in the exhaust stack about four times that of the atmosphere, the resulting *blowtorch effect* of the hot gases will quickly change a small leak into a gaping hole. The blast of escaping gases not only decreases the amount of energy available to the turbine wheel, but also constitutes a definite fire hazard.

Air Scoop

Beam a flashlight both into the air induction and intercooler intake scoops to see if any foreign objects are obstructing the flow of air through the ducts. Remove all accumulated foreign material. Objects such as newspapers, rags, name plates, bolts may cause serious damage to the impeller or intercooler during operation. Bulky material may restrict the flow of air to the turbosupercharger, thereby lowering the critical altitude of the power plant.

Induction System

Examine the ducts, joints, and gaskets of the airplane's induction system, from the air induction scoop to the internal blower, for any evidence of chafing, collapse, cracks, or other

signs of deterioration. Also examine the clamps on the neoprene fittings for security. Repair or replace any loose or faulty parts of the system.

Governor and Flexible Shaft

1. On installations equipped with the Type B electronic control system, inspect the connector on the turbosupercharger governor for tightness and proper insertion. The nut on the flexible shaft should be tight and *not* cross-threaded (Fig. 23).

2. In tropical areas or during very hot weather, disconnect the flexible shaft from the overspeed device every 25 hr and

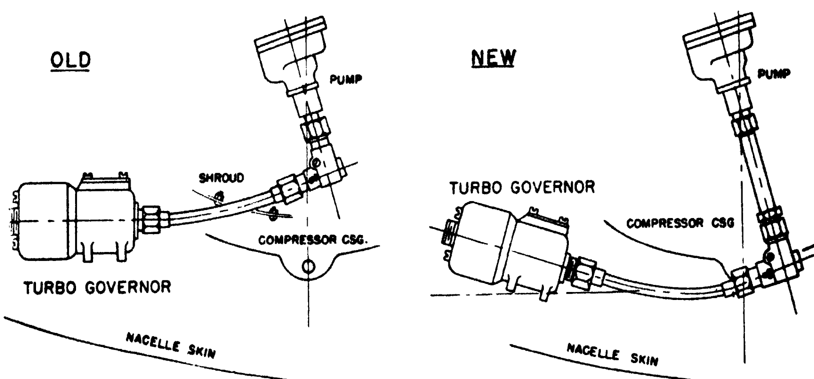


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Fig. 23. Turbo Governor Installation, Nacelle of B-29.

inspect the flexible shaft for lubrication and signs of failure. Use grease, if necessary. If the shaft is worn, or shows signs of failure, replace the complete drive. In normal operation, the inspection of the flexible shaft is taken care of during the 100-hr inspection.

Welded Bucket-wheel Rim

Remove the cooling cap and examine the bucket wheel for cracks in the weld along the rim of the wheel blank (Fig. 24). If cracks exceeding safe limits are found, replace the turbosupercharger.

Right-angle Adapter (B-11 and B-31 Only)

Give the right-angle adapter on the end of the rigid drive shaft one shot of grease. *Do not overgrease.* Greasing of this adapter is possible only on those B-11 and B-31 turbosuperchargers on which the right-angle adapter is mounted on the end of the rigid drive shaft that projects outside of the cooling shroud of the turbosupercharger.

Contamination of Oil

Draw off a small amount of oil from the turbosupercharger oil supply tank and inspect it for dirt or sediment. If the oil contains sediment, drain it out, flush the tank with flushing oil, and refill the tank with clean oil.

Total Side (Radial) Play of Rotor

Mount an indicator so that the pointer rests against the outside of the rim of the bucket wheel (Fig. 24). Move the

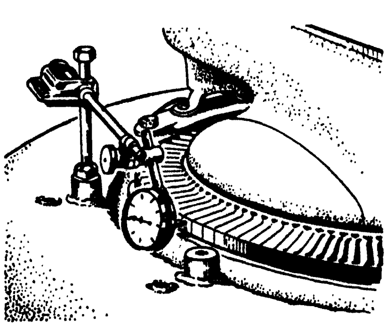

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Fig. 24. Checking Radial Play of Rotor.

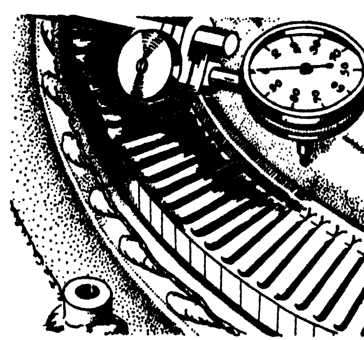

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Fig. 25. Checking Axial Play of Rotor.

bucket wheel from side to side to check the total side play of the rotor for bearing wear. If the total side play is greater than 0.003 in., replace the turbosupercharger.

Total End (Axial) Play of Rotor

Relocate the indicator so that the pointer rests on the flat rim of the wheel disk (Fig. 25). Move the bucket wheel up and down to check the total end play of the rotor for bearing

wear. If the total end play is greater than 0.009 in., replace the turbosupercharger.

Bucket-wheel Runout

1. Thoroughly scrape the rim of the bucket wheel until it is clean. Cleaning with a wire brush is not sufficient.

2. Mount the indicator so that the pointer rests on the flat rim of the wheel disk (Fig. 25).

3. Revolve the bucket wheel slowly by hand and locate the high and low points on the wheel rim. Also look for evidence of cracks or distortion. If the difference between the high and low points is greater than 0.005 in., or if any cracking is observed in the wheel disk, replace the turbosupercharger.

Lubrication and Inspection of Flexible Shaft

All installations with governors, overspeed warning devices, or tachometers driven by a flexible drive shaft between the oil pump of the turbosupercharger and the overspeed device, should have the flexible shaft inspected for possible failure. A broken shaft may lead to overspeeding of the rotor, with consequent stretching of the buckets or possible loss of the wheel.

1. Disconnect the flexible drive shaft at the overspeed device and spin the bucket wheel to see if the driving element of the shaft is rotating.

2. Pull out the flexible shaft and inspect it for wear. Lubricate with grease, if necessary. If the shaft is worn or shows signs of failure, replace the complete drive.

3. Replace the flexible shaft. After sliding it into the housing, press inward and turn until it slips into place and engages the drive connection on the turbosupercharger.

4. The couplings at both the oil pump and overspeed device should be secure after assembly and should not be cross-threaded.

Induction-system Pressure Test

1. Leaks in the induction system force the turbosupercharger to operate at a higher speed than normal to supply an excessive quantity of air. This may result either in the exhaust

back pressure becoming so high that normal engine power output can no longer be attained, or in the speed of the turbosupercharger rotor becoming so excessive that loss of turbine buckets or even the entire bucket wheel may result. The latter is especially likely to occur if the unit is not equipped with an overspeed governor. On installations incorporating an electronic or electric control system, overspeeding of the wheel is prevented; therefore, induction-system leaks will lower the critical altitude of the airplane.

2. Pressure test the induction system and repair or replace parts as necessary to eliminate leaks discovered in the induction system. The presence of a major leak in the intercooler core requires replacement of the intercooler.

Security of Turbosupercharger

With the aid of a flashlight, inspect for insecurity as much of the various assemblies of the turbosupercharger as possible. Loose mounting parts should be tightened.

Lubrication

The rotor of the turbosupercharger operates at extremely high speeds as compared with speeds normally encountered in other equipment (for one type, the speed is 21,300 rpm for a rated altitude of 25,000 ft). At this speed, the balls in the ball bearing are rotating about their own axes at approximately 60,000 rpm. Bearings capable of standing up under extreme conditions of speed must be of special design and are manufactured with precision. Therefore, no foreign matter should be allowed to enter the lubricating-oil system. Recommendations concerning type of oil to be used and the manner in which the turbosupercharger ought to be operated should be adopted.

Lubrication Precautions

1. Check the level of the oil in the supply tank prior to each take-off. If the oil level is not within one quarter of the top of the tank, add sufficient oil to maintain this level. Most turbosupercharger oil supply tanks have the filler neck located

on the side, so that when the level of the oil reaches the bottom of the lip of the filler neck, the oil is within one quarter of the top of the tank.

2. At least once during each 25-hr period of flight operation, drain some oil from the lowest point in the oil supply line leading to the turbosupercharger and inspect it for gritty dirt or other sediment that might damage high-speed bearings. If contamination is present, drain the supply tank and the oil lines, and refill the system with new oil. Clean oil may be used indefinitely.

3. No foreign material should be permitted to enter the turbosupercharger lubricating-oil supply. The smallest particle of dirt or grit in the oil might cause turbosupercharger bearing failure. The oil must be stored and carried in a closed container capable of protecting the contents from sand or other foreign matter.

Chapter XI

Wright Aeronautical Corporation

The source of power of the aircraft engine is the burning of the fuel-air mixture in the cylinders. The power developed by the engine varies primarily with the weight of the mixture that enters the cylinders: the greater the weight of the mixture burned the greater the power output.

The mixture consists of fuel and air, the fuel being metered to the air by the carburetor in a fixed ratio to form a combustible mixture. The carburetor can maintain the fuel flow to correspond to any mass of air flow. Since this is so, the problem of increasing power output resolves itself into increasing the mass air flow by compressing it so that a greater weight of mixture can be delivered to the cylinders.

The critical altitude of an engine (which is the highest altitude at which an engine will maintain rated power) is affected by the compressing ability of the supercharger. If the critical altitude of any engine is to be increased, the compressing ability of its supercharger must be increased proportionally. This increase may be obtained in various degrees depending upon the mechanical means used.

The single-speed one-stage internal supercharger (Fig. 1) in radial engines is composed of a centrifugal blower, or impeller, and a diffuser built in the engine between the carburetor and intake manifold. The power to drive the impeller is taken from the crankshaft, and, therefore, the impeller turns at a definite ratio of crankshaft speed. With this type of supercharger, additional supercharging at altitude may be obtained by increasing the capacity of the supercharger to an extent where its capacity is greater than is necessary to obtain maxi-

mum sea-level power. This would require operation at part-throttle at sea level, but would provide for a higher critical altitude. However, the relative gain in altitude obtained by this means is usually small.

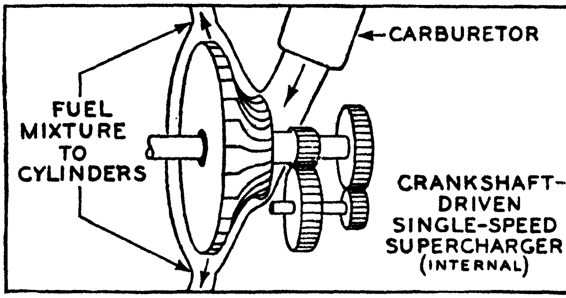

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Fig. 1. Crankshaft-driven Single-speed Supercharger.

Considerable gain in critical altitude can be obtained by incorporating a clutch system in the supercharger drive so that the ratio of impeller speed to the crankshaft speed can be increased above a given altitude, with a resulting increase in

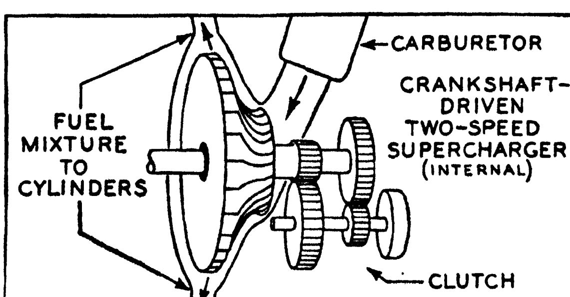

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Fig. 2. Crankshaft-driven Two-speed Supercharger.

supercharging. This type of supercharger, shown in Figure 2, is called a *single-stage two-speed supercharger*. The low speed is used at low altitudes and the high speed, at higher altitudes. In this case, the engine will have two critical altitudes, one for low-speed and one for high-speed ratio, and the latter will be at a much higher altitude.

Still more supercharging may be obtained by means of a two-stage internal supercharger in which the rarefied air is usually compressed in an auxiliary supercharger before it reaches the carburetor, and the mixture is later further compressed by a second supercharger before it enters the cylinders (Fig. 3). With this type of supercharger, the engine will

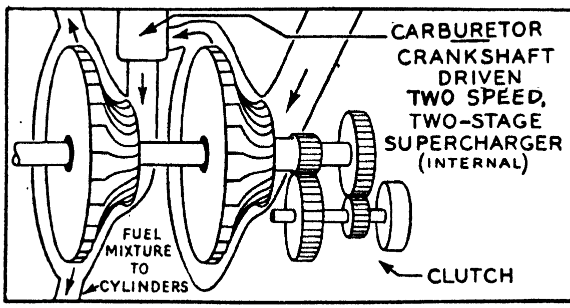

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Fig. 3. Crankshaft-driven Two-speed Two-stage Supercharger.

have three critical altitudes, since one of the superchargers incorporated in the engine will turn at one ratio to crankshaft speed and the other supercharger, at two ratios.

The superchargers mentioned thus far are internal—that is, they form part of the engine and are driven by engine power. In each case, the power developed at any altitude and the critical altitudes are dependent on the degree of supercharging and the amount of engine horsepower deducted to drive the supercharger.

Another method of supercharging, which has been particularly effective for high-altitude flying, is composed of the single-speed single-stage internal supercharger used in conjunction with an external exhaust-driven supercharger known as a *turbosupercharger*. This type of supercharger has an advantage over other types in that the speed of the turbosupercharger is infinitely variable and, therefore, only the necessary amount of supercharging is provided at all altitudes. With this combination of internal and external superchargers, there is only one critical altitude which is said to be the critical altitude of the turbosupercharger. It is interesting to note that

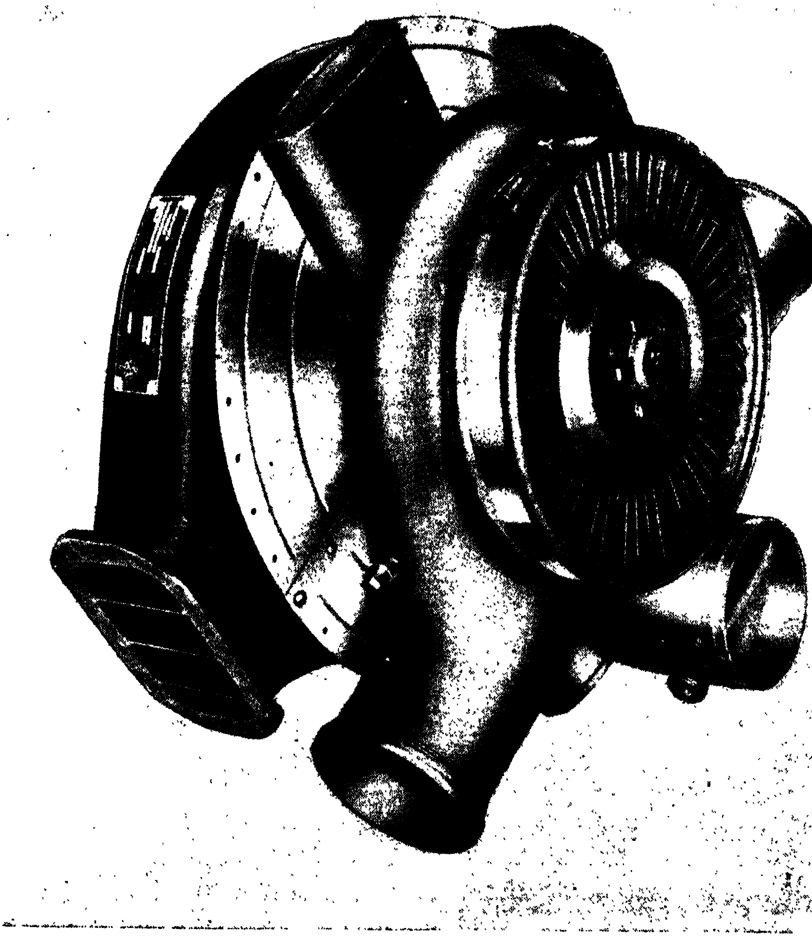

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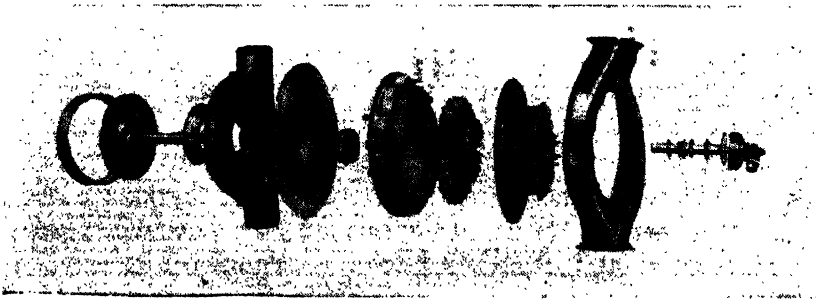
Fig. 4. Full View of Turbosupercharger Showing Turbine Wheel.



Courtesy of Wright Aeronautical Corporation

Fig. 5. Exploded View of Wright Turbosupercharger Model 800TSBA1.

the same manifold pressures at altitude are obtained with the single-stage internal supercharger and turbosupercharger combination as with the two-speed two-stage supercharger without the horsepower loss required to drive the internal auxiliary stage of the latter. In addition, with the turbosupercharger the aircraft is enabled to reach a higher altitude because the power available at any altitude is greater.

The operation, inspection, and maintenance of the Model 800TSBA1 turbosupercharger, Figures 4 and 5, designed and built by the Wright Aeronautical Corporation, will be explained in the remainder of this chapter.

Characteristics

The Wright turbosupercharger follows the conventional arrangement for turbosuperchargers. It is a separate unit consisting essentially of a centrifugal compressor secured to a turbine wheel which is driven by the exhaust gases of the engine. Therefore, it provides supercharging for the engine without using energy derived from the engine crankshaft.

The main *function* of the turbosupercharger is to supply air at sea-level atmospheric pressure to the engine-carburetor top deck when the internal engine-driven supercharger cannot obtain military or normal rated pressure for the altitude at which the engine is operating.

As to *rating:* when operating at 21,000 rpm, the turbosupercharger will maintain a supercharger outlet pressure of 31.67 in. of mercury at 23,100 ft standard altitude when used with an engine providing a rated air flow of 136 lb per min (1200 bhp at 0.113 lb per bhp per min).

When operating at 23,000 rpm, the turbosupercharger will maintain a supercharger outlet pressure of 31.67 in. of mercury at 26,800 ft standard altitude when used with an engine providing an air flow of 147 lb per min (1300 bhp at 0.113 lb per bhp per min).

The above rating is predicated upon a minimum exhaust-gas temperature of 1400° F and sufficient supercharging in the engine to provide rated and military air flow at standard sea-level pressure and 100° F carburetor air temperature.

For *lubrication,* the turbosupercharger is provided with an oil pump, and uses oil according to Specification AN-VV-0-446a.

An Eclipse or a Delco Remy turbosupercharger *pressure*

Courtesy of Wright Aeronautical Corporation

Fig. 6. Cutaway View of Wright Turbosupercharger Model 800TSBA1.

regulator is supplied with the turbosupercharger. Both regulators employ a device for controlling turbine speed.

The *compressor has two outlets* and a single inlet.

The *nozzle box* has two large inlet flanges and a single waste gate. An additional small inlet opening is provided for the sixth-cylinder exhaust.

A *tachometer drive* is provided.

Other characteristics are:

Weight of turbosupercharger	108 lb
Number of turbine wheel buckets	54
Number of nozzles on nozzle box	35
Turbine wheel diameter	11.44 in. (approx)
Impeller diam	12.5 in.
Number of blades on impeller	22

Description

For the purpose of description, the turbosupercharger may be thought of as being divided into two general sections: the exhaust turbine, or drive section; and the compressor, or driven section (Fig. 6).

Exhaust Unit

The engine exhaust gases are conducted by means of ducts to the turbosupercharger exhaust manifold, commonly called the *nozzle box* (A). The nozzle box may have one, two, or three inlet connections (B), although the first Wright production Model 800TSBA1 has three. The exhaust gases are then directed through nozzles (C) so that they impinge against the buckets (D) of the turbine wheel, thereby driving the wheel. There are 35 nozzles in all, contained in a cast ring welded into the nozzle box.

The waste gate which regulates the flow of exhaust gases through the turbine is housed in an elbow (E) which is welded into the nozzle box. The waste gate is merely a butterfly valve through which exhaust gases can be expelled overboard.

For measuring the exhaust pressure, two pressure connections are furnished, one (F) located near each of the large exhaust inlets of the nozzle box.

One of the main features of the Wright turbosupercharger

is the design of turbine wheel buckets. Fifty-four buckets are bolted between two hubs (*G*) which are in turn bolted to the impeller shaft (*H*). The ability of these buckets to withstand the extreme temperatures of the exhaust gases determines to a major degree the speed at which the turbine wheel can be safely turned. Therefore, much care has been given to selecting material with high heat-resistant qualities and to designing buckets to provide adequate means of cooling.

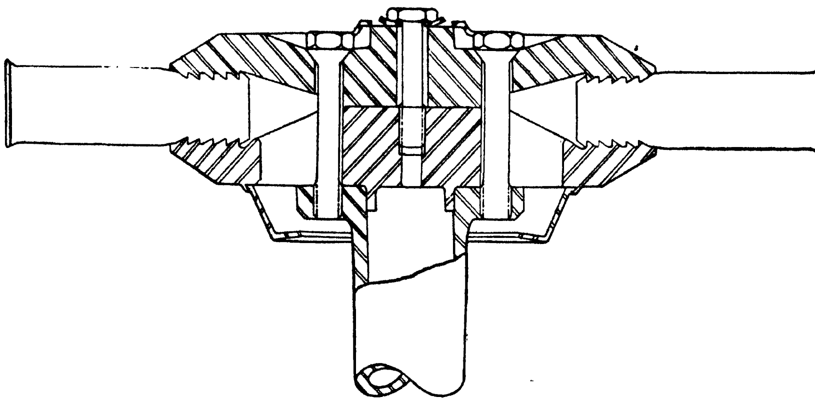
An examination of (*D*) of Figure 6 will show how the buckets are designed for cooling purposes. They are hollow throughout to allow a stream of cooling air to pass through their entire length. As the turbine wheel rotates, the air in the buckets is forced out of the outer end of each bucket owing to centrifugal force, thus providing an inherent pumping action to the turbine wheel. The outer parts of the buckets are concave on one side and convex on the other. It is on the concave surfaces that the exhaust gases of the nozzle box impinge to drive the turbine wheel. As the hot gases strike the concave surfaces, the inner air cools the material of the buckets to a relatively low temperature despite the high temperature of the exhaust gases.

To prevent the buckets from pulling out when the turbine wheel turns at high speed, a locking system has been devised. The inner ends of the buckets are flat on two sides (for placing one bucket close against the other) and the other two sides contain teeth (for mating the buckets with the concentric teeth of the hubs). When the two hubs are bolted by the five outer hub bolts and the one center bolt, the buckets and hubs are securely locked together (Fig. 7).

The source of the cooling air is an inlet (*I*) which is integral with a bell-shaped support (*J*). This support is between the nozzle box and the compressor units, and is spoken of as the *nozzle box support*. The cooling air enters the inlet and is led through the bore of the nozzle box to openings in the inner turbine-wheel hub which allows the air to enter the buckets.

Another means used in the turbosupercharger to divert heat is the heat shield. Two heat shields are used, one on each

side of the nozzle box. The purpose of the two-piece shield (K) which surrounds the nozzle-box support is to take the heat away from the compressor units, and the purpose of the other heat shield, (L), which lies between the nozzle box and



Courtesy of Wright Aeronautical Corporation

Fig. 7. Schematic Diagram of Turbine Wheel.

turbine wheel, is to divert some of the heat from the cooling air on its way to the buckets.

It is interesting to note that all parts of the exhaust section are made of stainless steel with the exception of the turbine wheel buckets. The latter are made of a special alloy, Westinghouse K42B, which is composed principally of nickel, cobalt, and chromium.

Compressor Unit

Atmospheric air, which is to be compressed in the turbosupercharger and delivered to the engine carburetor top deck, enters the turbosupercharger impeller (M) after having passed through an air scoop and the supercharger inlet housing (N). From the impeller, the air, which has greatly increased in speed, flows between the vanes of the diffuser (O) to the supercharger outlet manifold (P). It then exits through the two manifold outlets into ducts which convey the air through the intercooler to the engine.

Besides serving as an inlet for atmospheric air, the super-

charger inlet housing contains passages for incoming lubricating oil (Q), scavenge oil, and pressure-gauge oil (R). Tubes are pressed in the lubricating and scavenge passages, with 45-deg elbows screwed in as connections. Pressure-gauge oil is led through a hollow dowel pin from the oil pump (S).

The impeller is mounted on the tapered section of the impeller shaft (H) which is bolted to the turbine wheel, and secured by a nut. In order that the impeller may turn with the turbine wheel, it is necessary that the former be securely attached on the shaft taper; this is insured by tightening the nut to 2000 to 2100 in.-lb. Bronze rings (T), one on each side of the impeller nut, and a carburized washer (U) absorb the thrust of the shaft.

The diffuser (O) is integral with the impeller shaft housing (V). Enclosed in a retainer attached to the exhaust end of the housing is a device for dampening and limiting the radial movement of the impeller shaft. This device is composed of a sintered bronze bearing (W), a floating steel cage, a bronze plate, and six springs and sleeves. The sintered bronze bearing fits in the bore of the steel cage with clearance between it and the impeller shaft. Any radial movement of the shaft will then cause the cage to move with respect to the bronze plate. The friction at this point thus tends to dampen any vibration of the impeller shaft. Furthermore, the cage serves to limit the movement of the impeller shaft during violent maneuvers of the airplane by bottoming on the sleeve.

The compressor unit is bolted to the exhaust section by means of 12 bolts (X) which lock the nozzle-box support (J) against the impeller shaft housing.

The turbosupercharger is supplied with a tachometer drive (Y) which is secured to the oil pump by means of screws. The tachometer drive mechanism as well as the oil pump (S) is driven by a slender shaft (Z) that fits into a coupling and support in the impeller shaft. Thus, the oil pump and tachometer drive shaft turns with the impeller shaft. The tachometer driven shaft is connected by means of a cable to the turbosupercharger overspeed governor when an Eclipse turbosupercharger regulator is used.

Lubrication

The Wright turbosupercharger uses engine oil for lubricating purposes. The oil supply is obtained from either the engine oil tank or a separate tank. Depending on the installation, an oil filter may or may not be placed in the oil line leading to the turbosupercharger.

Path of Lubricating Oil

The turbosupercharger is equipped with a pressure pump and internal piping in the compressor (supercharger) inlet housing. External inlet and scavenge piping lines lead from the oil supply to the inlet housing.

The lubricating oil enters the inlet connection and is carried by internal passages to the oil pressure pump. After the oil passes the pressure gears, it follows two paths (Fig. 6).

The larger quantity of oil flows along the inside and outside of the impeller shaft support which is located in the impeller shaft. In this passage, the oil lubricates the oil pump. tachometer drive gear, and the two bushings pressed in the impeller shaft. The oil then circulates back to an outlet passage in the supercharger inlet housing.

Oil from the pump also circulates to the tachometer drive gears. A small passage in the oil pump housing leads the scavenge oil to the scavenge line in the supercharging inlet housing.

Oil-pressure Relief Valve

An oil-pressure relief valve is provided in the oil pump. If the pressure of the oil after passing the pressure gears exceeds a predetermined maximum, a spring in the valve compresses and allows the oil to pass back into the inlet section of the oil pump.

Pressure-gauge Connection

In the compressor inlet-housing, an oil-pressure connection and passages leading to the oil pump are provided.

Principles of Operation

In order to explain how each unit used in the turbosupercharger-engine combination affects the mass air flow of the power plant, the cycle through which a particle of air goes will be traced (Fig. 8).

Assume the airplane to be flying at 25,000 ft with the engine developing military rated speed. The particle of air is at

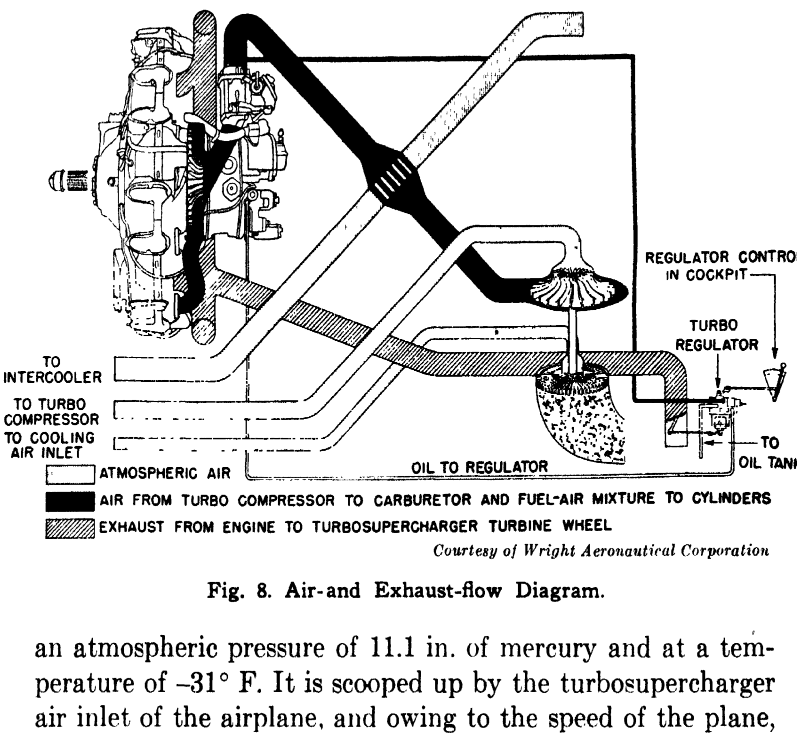


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Fig. 8. Air-and Exhaust-flow Diagram.

an atmospheric pressure of 11.1 in. of mercury and at a temperature of –31° F. It is scooped up by the turbosupercharger air inlet of the airplane, and owing to the speed of the plane, the particle is rammed into the inlet of the turbine-driven compressor. There it is accelerated by the rotating vanes of the impeller, and acquires a high velocity by the time it leaves the tips of the impeller. The impeller has done a certain amount of work and the particle has absorbed this work in the form of energy which it now possesses by virtue of its speed.

The speeding particle then enters the stationary vanes of the diffuser and is slowed down until most of the energy it

contains from its speed is converted to energy that is expressed in the form of an increase in pressure. This pressure will be about sea-level pressure. The remainder of the velocity, or kinetic energy, that is not converted to pressure is expressed in the form of heat. The temperature will now be about 200° F.

Since air is more dense when it is cold, more air can be pumped through the engine if it is cooled after leaving the compressor. The hot particle is, therefore, passed through an intercooler, which reduces its temperature to about 86° F.

There is a slight pressure drop as the air passes through the intercooler, owing to the resistance of the cooling tubes to the air flow. The air enters the carburetor at approximately sea-level pressure and a temperature of about 86° F.

As the air particle passes through the carburetor, it is measured together with all the other particles as a rate of flow, and the proper amount of fuel is injected into the air stream.

The particle of air, which now has a bit of fuel along with it, enters the gear-driven supercharger of the engine, and is further compressed in the same manner that it was in the turbine-driven compressor. The pressure thus obtained, known as *manifold pressure,* is, for any given engine rpm, the controlling factor which decides how much air (or combustive mixture) is to pass into the combustion chamber and, consequently, how much power is to be developed at that particular engine rpm.

The particle is now one of many which make up a very dense, combustive mixture of air and fuel. In the cylinder, the heat energy of the fuel is released. The burning, expanding mixture then does work on the piston, giving up a little more than a third of its energy, and passes out through the exhaust port into the engine exhaust manifold and back to the turbine.

By allowing certain amounts of exhaust gas to pass overboard through the waste gate, the pressure in the nozzle box can be maintained at any pressure desired. This is done by the pressure regulator.

The particle passes through the nozzles of the turbine and impinges on the turbine blades, thus converting much of its

stored energy into useful mechanical work that can be used to drive the turbosupercharger impeller.

One important fact must be reiterated in regard to the function of the regulator: the pilot does not, cannot, and should not attempt to operate the waste gate. This is done solely by the regulator as it sets up the pressure called for by the pilot in his manipulation of the regulator control lever.

Service Inspection and Maintenance

Since the turbosupercharger operates at extremely high temperatures and speeds, great care should be exercised in all inspections. It is recommended that the same periods of inspection be observed as those assigned for engines—that is, the daily, 25–30-, 50–60-, and 90–100-hr intervals. These periodic inspections should include, besides the turbosupercharger unit, the ducts, oil lines, carburetor deck line or compressor discharge line, the regulator and operation of the waste gate, the overspeed governor, and the intercooler.

Daily

1. Check to see that an ample supply of lubricating oil is available and that the oil drain valve is closed and saftied.

2. Inspect visually the turbine wheel shroud, exhaust hood, and the turbine wheel, especially the buckets. Spin the wheel to insure that it has freedom of movement. If the wheel does not turn freely and no obvious cause can be found and corrected, the unit must be removed and disassembled.

3. Inspect for oil leakage. Should there be evidence of an oil leak between the tachometer drive and oil pump housings, the tachometer may be removed and a new gasket installed.

4. Give a general visual inspection of all duct work, intercoolers, and supercharger waste gate. If excessive carbon has caused the waste gate to become tight in its bushing supports, the gate should be removed. Carbon can then be taken off the bushings with emery cloth and a hand scraper. After installing the waste gate, it will be necessary to readjust the turbosupercharger regulator control so that the gate is operating properly.

Twenty-five-to-thirty-hour Inspection

In addition to items listed under *Daily* inspection, the following should be checked:

1. See that the waste gate works freely. Noticeable friction or binding is evidence of warpage. A warped waste gate should be replaced.

2. Using a feeler gauge, check the clearance of the waste gate in the closed position. This clearance should be 0.005 to 0.030 in. If the clearance is not within the required limits, refer to paragraph (1) above and (4) under *Daily* (Fig. 9).

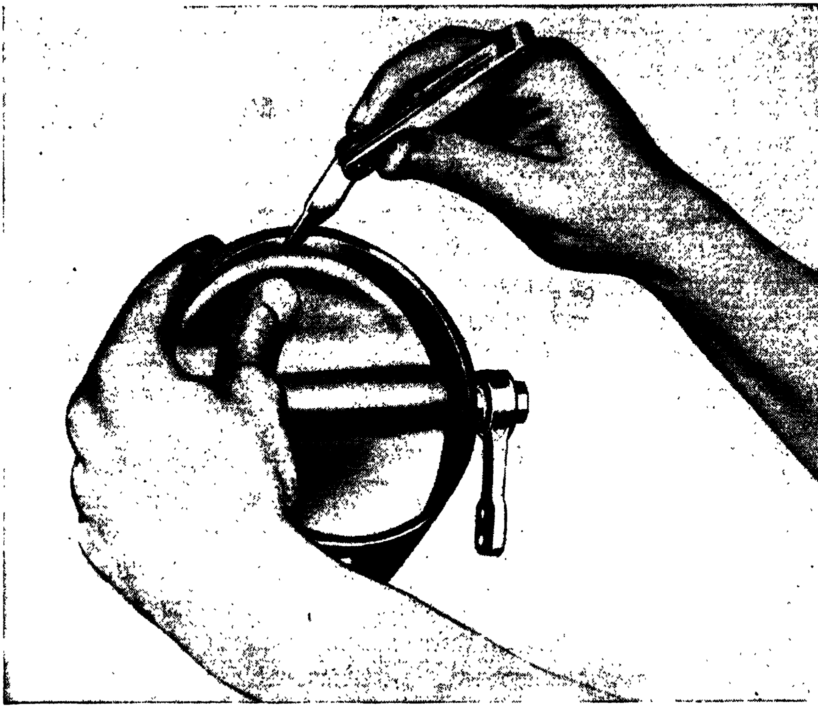
3. Examine the exhaust hood for cracks. If cracks are found the hood should be replaced.

Fifty-to-sixty-hour Inspection

In addition to items listed under *Daily* and *Twenty-five-to-thirty-hour* inspections, the following should be examined:

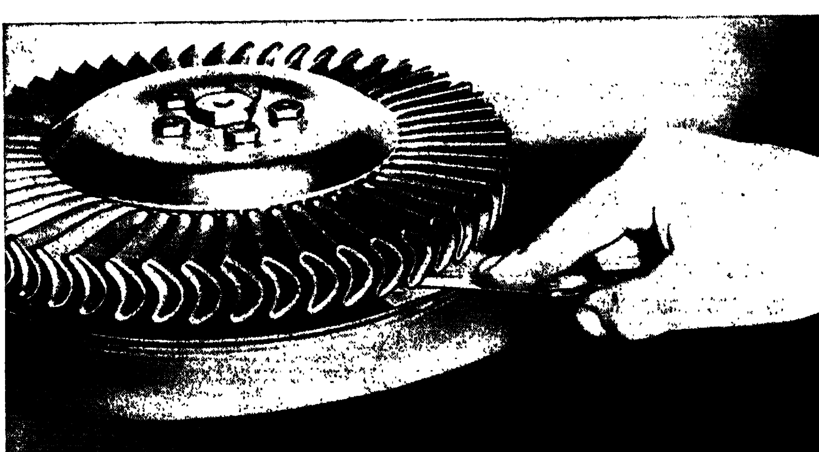
1. Check the end clearance between the turbine wheel buckets and shroud. This clearance should be 0.1-in to 0.15-in. If the clearance is not within the required limits, the difficulty may be caused by a warped shroud. The shroud may be repaired and reinstalled, or replaced by a new shroud. If the incorrect clearance is due to the turbine wheel, the turbosupercharger must be disassembled and inspected.

2. Remove the shroud and check the clearance between the turbine wheel buckets and nozzle box. If the clearance is not within the required limits, the turbosupercharger must be removed and given a complete overhaul. To measure the nozzle-box-to-bucket clearance, proceed as follows: Hold a plate of predetermined thickness against the nozzle box and insert feeler stock at a 30-deg angle under each bucket, turning the wheel until the clearance of the bucket closest to the nozzle box at this point has been determined; again turning the wheel, take the clearance between this bucket and the nozzle box at several points approximately equally distant around the box (Fig. 10). The lowest reading will give the nozzle-box-to-bucket clearance. If the turbosupercharger is installed with the turbine wheel facing up, the clearance should



Courtesy of Wright Aeronautical Corporation

Fig. 9. Measuring Wastegate Clearance.



Courtesy of Wright Aeronautical Corporation

Fig. 10. Measuring Nozzle Box Clearance.

be 0.16 in. to 0.24 in.; if installed with turbine wheel facing down, the clearance should be 0.17 in. to 0.25 in.

3. Drain off a sample of oil from the supply and remove the strainer, if used. Examine both for presence of foreign matter. If the oil contains sediment or the like, clean the tank and fill it with new oil. If the same oil supply is used for engine and turbosupercharger, and any internal trouble is experienced which has caused metal particles to accumulate on the engine oil strainer, it will be necessary to disassemble and clean the entire supercharger.

4. Inspect all connection lines for leaks and wear. Repair if necessary.

Ninety-to-one-hundred-hour Inspection

At this time, the whole turbosupercharger system should be checked. The fairings should be removed, if necessary, to obtain access to the turbosupercharger. All air ducts should be inspected for cracks. Oil lines should be removed from the turbosupercharger regulator to permit drainage, so that a check can be made of the operation of the waste gate. The waste-gate lever should close against the stop and open to 60 deg.

Below are listed parts that must be inspected at the 90–100-hr period.

1. *Mounting* for: cracks, chafing, rust, and the condition of bolts and clamps.

2. *Turbine wheel* for: burnt or cracked buckets, rust or cracks of the upper hub, clearance between wheel and nozzle box, and clearance between wheel and shroud.

3. *Shroud* for: condition of bolts, bends, dents, or cracks.

4. *Nozzle box* for: warpage, dents, cracks, and tightness of connections; waste-gate warpage and operation, cleanness of bushings, clearance, and tightness; leakage or chafing of pressure connections.

5. *Outer heat shields* for: cracks, dents, bends, and condition of the screws.

6. *Supercharger outlet manifold* for: leaks or cracks in outlet connections, cracks, and dents.

7. *Supercharger inlet housing* for: cracks, dents, leaks, or

chafing of oil and gauge connections; and condition of the bolts and nuts.

8. *Impeller blades,* as seen through inlet housing (if accessible), for: cracks or chips.

9. *Oil pump* for: cracks, oil leakage, and condition of the screws.

10. *Tachometer drive housing* for: cracks, oil leakage, and condition of the screws.

11. *Compressor ducts* for: cracks and leakage.

12. *Supercharger regulator* for: operation and travel of linkages and levers, and cleanliness and leakages of oil.

13. *Gauge lines* for: deterioration, chafing, and leakage.

Chapter XII

Testing of Turbosuperchargers[1]

Description of Test Equipment

The testing of exhaust-gas turbochargers, or turbosuperchargers, falls into two categories: first, the experimental testing in connection with the development or refinement of a design; and second, the testing of each production unit after assembly to certify its correct operation. Of the latter type of test, only a short description is necessary. The production turbosupercharger unit is usually given two, sometimes more, runs. The initial, run-in, or green run, fulfills two purposes. (1) By a gradual warm-up from idling to full speed, it insures the proper seating of mating parts. (2) It serves as a check on performance, clearances, rotor balance, lubrication system operation, and so forth. After an initial run, a unit is usually completely disassembled and its parts inspected for damage or wear. If none has occurred, the unit is then reassembled and tested a second time. This operation check is often only a 15-min warm-up from idling to full speed for the purpose of verifying correct reassembly of parts; from this test the unit goes directly to the purchaser. If, however, after the initial run, any part is found to be damaged or excessively worn, that part is replaced, the cause of the damage ascertained and eliminated, and the unit is reassembled and retested. The procedure followed subsequent to this penalty run is the same as that following an initial run.

Generally, one unit out of every lot is given a calibration run to check conformance of compressor and turbine design

[1] Prepared by D. M. Shackelford, Research Department, American Locomotive Company.

to standard. Of this type of test, more will be said later in the discussion of experimental testing.

Steam Pressure System for Production Testing

Since little instrumentation is necessary, the test-stand design for production testing is expressive of two cardinal principles: ease and rapidity of unit installation and removal, and economy. Steam is usually used to drive the units, because today large industrial plants are their own producers of high-pressure high-temperature steam Figure 1 shows a steam-driven test rig for production use. To speed up installation. connections between the unit and the stand ductwork, wherever possible, are two-piece or hinged clamps fitting around a flange on the turbosupercharger and a mating flange on the ductwork (Fig. 2). Sliding joints in the ductwork permit easy alignment of elements, and hydraulic lifts and sliding platforms reduce fatigue of test personnel. This test stand is designed for the location of the control room to the left of the test cell; therefore, the moving elements of the turbosupercharger are arranged to rotate in a vertical plane, so that in case of accident, the wheel and impeller sections will be thrown off away from the control room. Were the control room to be located above or below the test cell, the unit would be so installed that the turbine would rotate in a horizontal plane. This principle also applies to the design of experimental test cells.

Experimental Testing

Experimental, or developmental, testing is carried on in four areas. First, the test cell, which is used for the testing of a design until the turbosupercharger, in and of itself, is a satisfactory performer. Second, the full-scale test stand, where the turbosupercharger is tested in combination with accessories and with the engine or engines it was designed to supercharge. Third, the operating field, where the power plant is mated to the aircraft, locomotive, or ship, and the combination is tested under conditions at which it was designed to operate. Fourth, the consumer, who in reality is a test area rather than an ex-

perimental area. Often reports are received of troubles experienced in the field, troubles which must be referred back to the test cell for solution. No design is of proven excellence before it has been tested in this widest of all areas. The use of turbosuperchargers in the field is, however, outside the scope of this chapter; only the first three areas will be discussed.

REFERENCES TO PARTS IN ILLUSTRATIONS OF CHAPTER 12

1—Steam-pressure Regulating Valve
2—Oil Temperature Regulator, Heater
3—Oil Flow Regulator
4—Tachometer
5—Oil Pressure Gauge
6—Steam Pressure Gauge
7—Potentiometer for Low-temperature Measurement
8-9—H_2O Manometer
10—H_2O Manometer
11—Hg Manometer
12—Hg Barometer
13—Gauge for Oil Scale
14—Oil Flow Rotometer
15—Vibration Analyzer
16—Thermocouple, Oil Temperature into Turbosupercharger
17—Thermocouple, Oil Temperature out of Turbosupercharger
18—Oil Filter
19—Oil Cooler
20—Turbosupercharger
21—Air Filter
22—Air-flow Measuring Orifice, Compressor
23—Hydraulically Operated Sliding Platform
24—Hydraulically Operated Lift on Overhead Track
25—Steam-line Rotating Joint
26—Compressor-inlet Air Throttle
27—Adjustable Smooth-approach Orifice
28—Compressor-outlet Air Throttle
29—Stand Potentiometer
30-31—Inclinometer Connections
32—Air Bleed for Dew-point Indicator
33—Cooling Air Throttle
34-35—H_2O Manometer
36—Electric Igniter, self-retracting
37—Combustion Chamber
38—Fuel Nozzle
39—Fuel Pump
40—Fuel Tank
41—Fuel Control Valve
42—Fuel Pressure
43—Fuel Flow Rotometer
44—Hg Manometer
45—Manometer, H_2O or Hg
46—Potentiometer for High-temperature Measurement
47—Suction Blower
48—Air Control Valve
49—Air Filter in Cell Wall
50—Hg Manometer
51—Hg Manometer
52—H_2O Manometer
53-54—H_2O Manometer
55—H_2O Manometer
56—Intercooler
57—Air Dryer
58—Air Blower
59—Expansion Turbine
60—Air Brake
61—Air-brake Inlet Throttle
62—Air-brake Outlet Throttle
63—H_2O Manometer
64—Hg Manometer
65—Hg Manometer
66—Hg Manometer
67—Furnace Air Throttle
68-69—H_2O Manometer
70—Adjustable Smooth-approach Orifice
71—Air-brake Oil Pressure
72—Tachometer
73—Water to Intercooler
74-75—H_2O Manometer
76—Starting Blower
77-86—Shut-off Valves
87—Hand Oil Pump

A-E—Duct Connections
(a) Sliding Joints, Telescoping
(b) Split Ring or Hinged Clamps
(c) Seamless-bellows Expansion Joints
(d) Bolted Flanges and Gaskets

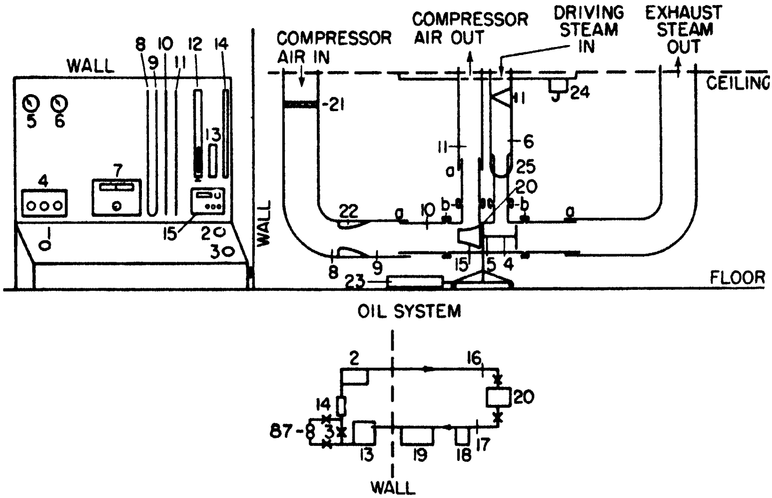


Fig. 1. Steam Pressure System for Production Testing. Above, left, control board; above, right, test cell; below, oil system.

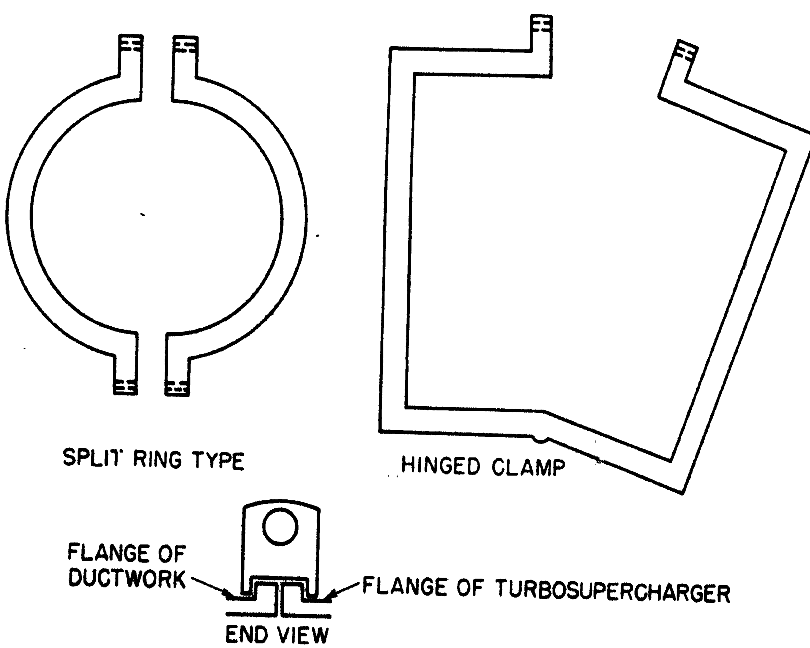
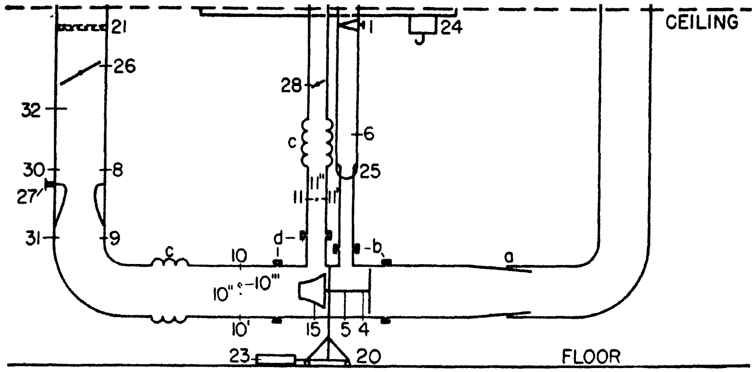
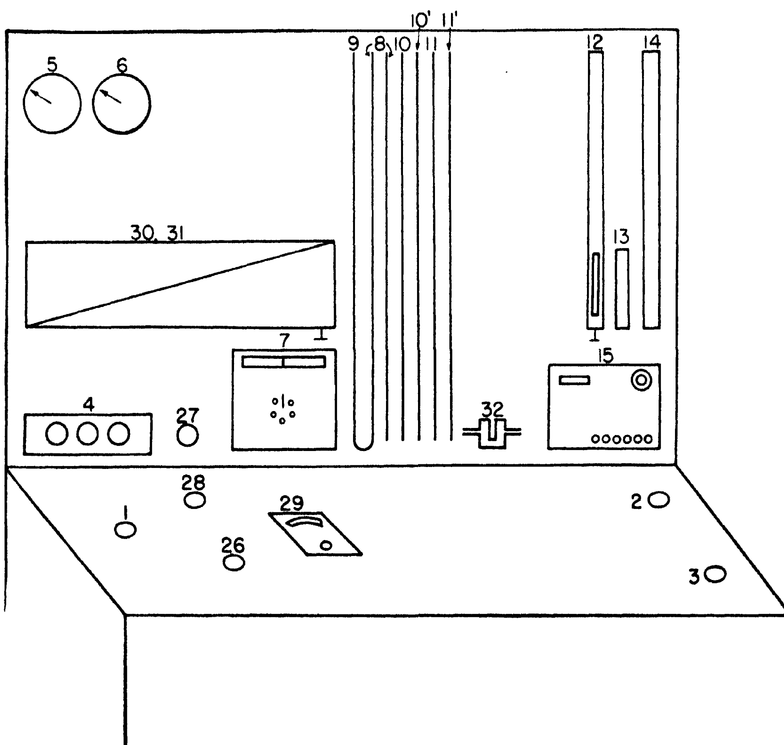


Fig. 2. Ductwork Clamps.





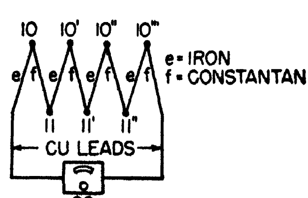


Fig. 3. Steam Pressure System for Experimental Testing. Above, test cell; below, control board; at right, average compressor-rise thermocouple system.

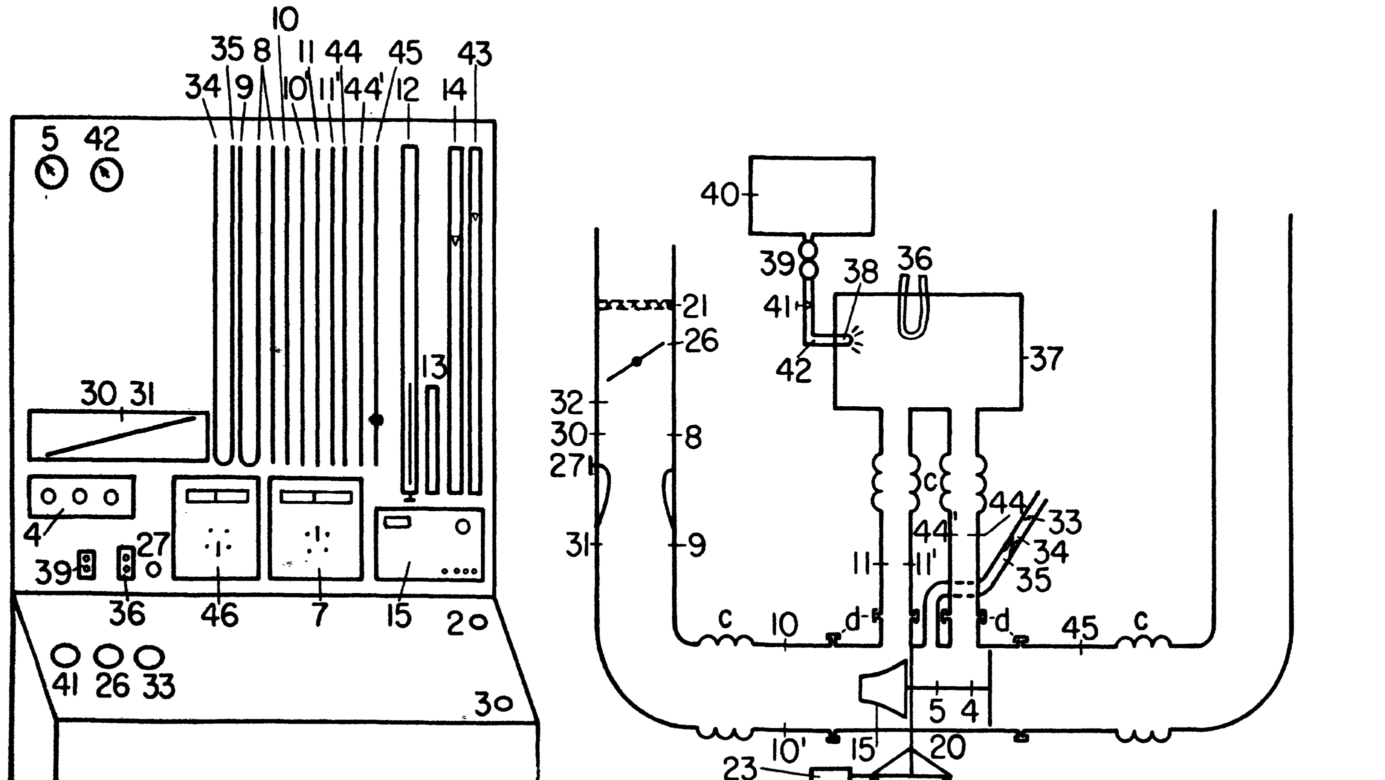


Fig. 4. Regenerative Test Stand. At left, control board; at right, test cell.

Experimental test cells show a wide variety of design depending upon the type of test to be run. For the testing of lubrication systems, a setup such as that used for production testing (Fig. 1) is sufficient. The supply of oil to the turbo can be throttled by adjusting the setting of valve (3), and lubricating characteristics of different grades of oil at various temperatures, flow rates, and pressures can be studied. Changes in oil-seal design, bearing materials, and oil-pump relief-valve spring settings can be investigated here as thoroughly as on a stand of more complex design, with the advantages of easy unit installation, simplicity of control, and cheap motive power.

Steam Pressure System for Experimental Testing

Another type of test stand that can utilize steam for driving the turbosupercharger is that shown in Figure 3. This setup is used for compressor testing, and must have a leak-free compressor ducting system. Therefore, bellows expansion joints replace the sliding joints of Figure 1, and bolted mating flanges with gaskets replace the hinged clamps in the compressor system. The addition of an inclinometer for the rapid determination of load variation, an accurate air-flow measuring device, controls for regulating air flow, and more sensitive pressure and temperature instrumentation make it possible to determine accurately compressor characteristics, such as efficiency and range.

Regenerative Test Equipment

The type of test stand most widely used for turbosupercharger testing is known as *regenerative,* in that the air from the compressor is led into a combustion chamber, mixed with fuel oil, and burned to produce hot gas for driving the turbine. Such a system is shown diagrammatically in Figure 4. Here, expansion joints are introduced into the turbine ductwork and the compressor ductwork, and the control and instrumentation systems are expanded to permit regulation of all components. On such a test stand, the effect of temperature and pressure on the turbine system, the effect of cooling air, and the corrosive effects of salt water, dirt, and fly ash can be

studied. This type of test stand is used for turbosupercharger calibration.

Ejector Test Equipment

For the study of nozzle-box and turbine-bucket design, the regenerative test stand is often used with the addition of an exhaust-gas analyzer so that the properties of the driving gas may be determined. Preferable, however, is the use of a steam-ejector setup such as is shown in Figure 5. Here, the use of

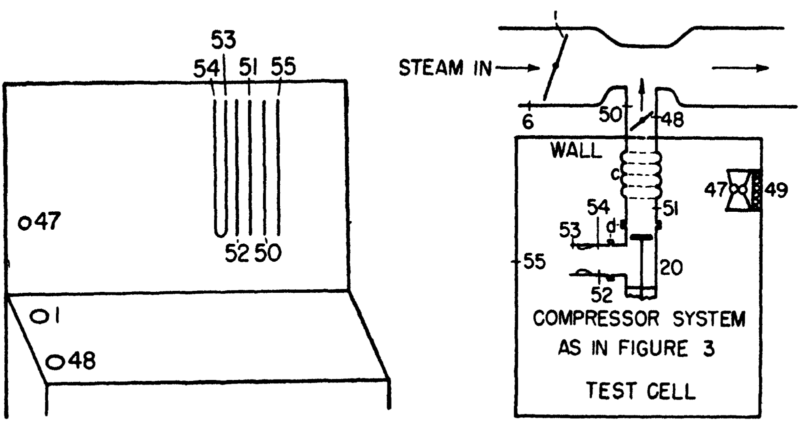


Fig. 5. Ejector Portion of Test Stand. At left, control board; at right, test cell.

clean ambient air (drawn by cell blowers into the test cell through filters in the cell walls) as the motive fluid permits accurate analysis. An increase in the flow of steam past the venturi section decreases the pressure on the exhaust side of the turbine, and increasing the flow of air through the turbine increases the pressure drop. Throttling the air flow at the inlet to the compressor permits load and speed regulation at all pressure drops.

Altitude Testing

The testing of turbosuperchargers under simulated altitude conditions introduces a new problem, that of reducing the moisture content, temperature, and pressure of the air for the compressor inlet. Since the drying of large volumes of air necessitates the use of expensive equipment, a closed compressor

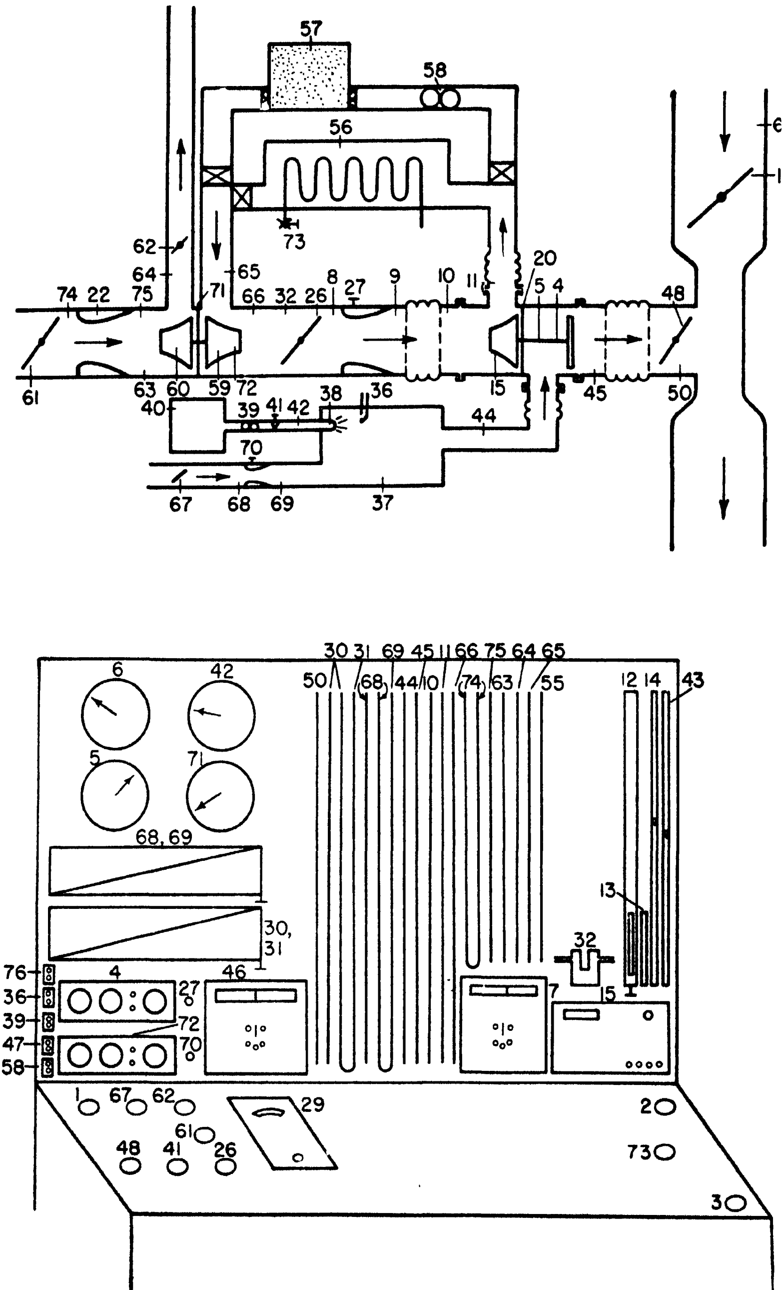


Fig. 6. Altitude Test Stand. Above, test cell; below, control board.

system is employed in order that the dehydrated air may be used repeatedly. Intercoolers after the compressor outlet remove the heat added to the air during compression, and an expansion turbine before the compressor inlet reduces the temperature and pressure to the desired level. An air brake with variable air flow, used in connection with the expansion turbine, permits load variation on the expansion turbine (Fig. 6). The turbine may be either steam-driven (Fig. 3) or steam ejector-driven (Fig. 5) or, preferably, a combination of steam ejectors and an unpressurized combustion chamber, as indicated. This permits the control of turbine inlet temperature while maintaining pressures comparable to those experienced in operation at altitude.

Convertible Test-stand Equipment

For the plant that is unrestricted by the limitations imposed by lack of space and capital, each type of test stand has its place, and development can best be carried out by using each for the purpose for which it was intended. Most companies, however, do not find themselves in this fortunate situation, and are, therefore, forced to compromise. For such companies, a sketch is included (Fig. 7) of a convertible test stand, which, except for a few changes of short sections of ductwork, can be made to do the job of any of the five types of test stand described above. And with the provision of short pieces of adapter ductwork, the stand could be designed to take care of turbos of varying diameters, sizes, angles, and numbers of inlets and outlets.

With so much of the development work done in the test cell, very little testing remains to be done on the full-scale stand and in the aircraft, locomotive, or ship. The full-scale stand takes care of the power-plant calibration and settles questions concerning regulator and master control settings, and the corrosive effect of engine exhaust gas. The latter is often more severe than that experienced on a regenerative stand since there, for the sake of economy, fuel oil rather than gasoline or Diesel oil is usually used. Testing in the vehicle in-

vestigates the effects of rate of climb, changing directions and attitudes of flight, and various flame-suppressant measures.

With the advent of the gas turbine, another test area has come to the fore—that is, the flying test bed. Two papers read

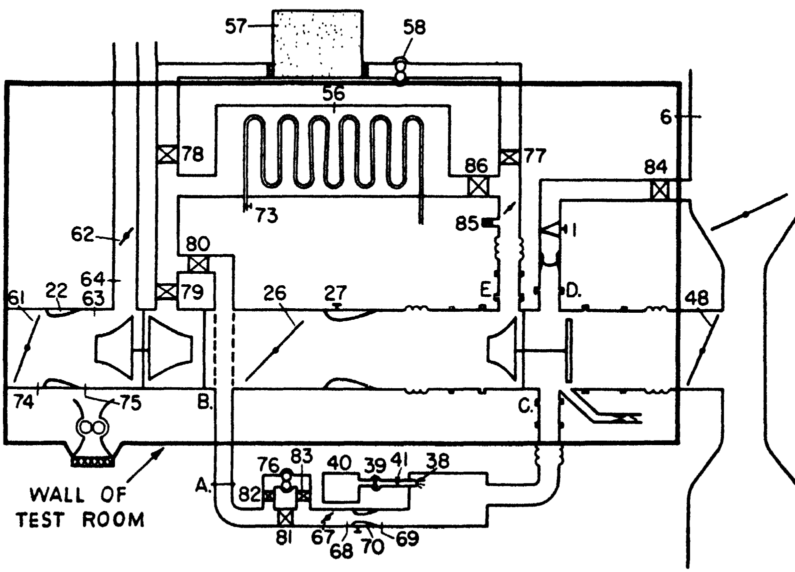


Fig. 7. Convertible Test Stand.

before the November 1945 meeting of the A.S.M.E. give detailed descriptions of this type of testing.

Test Personnel

The testing of experimental exhaust-gas turbosuperchargers can best be explained with reference to the last described test stand, since the very complexity of its equipment and instrumentation facilitates the adaptation of the instructions which follow to any simpler type of stand. This test stand (control panel as in Fig. 6) is designed for operation by three persons: a tester, located at the extreme left of the control panel, who is responsible for the condition and maintenance of the test equipment and for the operation of the turbosupercharger unit in accordance with the test engineer's in-

structions; the test observer, at the extreme right, who is responsible for the neatness, completeness, and accuracy of the test log, and who assists the tester and is under his supervision; and the test engineer, who is responsible for the safety of the turbosupercharger and the engineering data derived from the test. Instructions regarding test limits and test points to be run are issued by him.

Preparation

Preparation for the test of a turbosupercharger begins as soon as the unit that has been tested previously is removed from the stand. The test engineer should at that time (preferably by means of a schedule posted in the control room) advise the tester what unit is to be tested next, and the estimated time of arrival (E.T.A.) of the unit. Such a schedule might have the following form:

Unit No.	Type of Test	Size	No. Inlets	Inlets	No. Outlets	Outlets	E.T.A
162	Comp. calib. steam	2′d	3	45°	2	90°	1–11–44 8:30 PM
254	Turbine calib. ejectors	3′d	4	30°	2	90°	1–15–44 9:00 AM

Thus, the tester may schedule such repairs as are necessary, change ductwork before the arrival of the turbosupercharger, and utilize to best advantage the stand's idle time.

If the stand is designed for testing turbosuperchargers of various designs, many different sets of adapter ducts are necessary. To avoid confusion, these ducts should be labelled plainly, either by name or by a prearranged number system. such as:

No. 1, 1 outlet, 3″ diam, 90 deg	No. 2, 2 outlets, 3″ diam, 90 deg
No. 3, 1 outlet, 4″ diam, 90 deg	No. 4, 2 outlets, 4″ diam, 90 deg
No. 5, 1 inlet, 4″ diam, 90 deg	No. 6, 2 inlets, 4″ diam, 45 deg
No. 7, 3 outlets, 2½″ diam, 30 deg	No. 8, 4 outlets, 3″ diam, 10 deg

Also, crews are helped in installing ductwork by a line on the adapting portion of the ductwork that coincides with a similar line on the stationary portion. The use of different colors on oil, air, fuel, and water lines and valves; metal iden-

tification tags referring to similarly numbered manometers; and a posted chart of the stand saves considerable time, and often prevents damage to a unit or the accumulation of worthless data.

The maintenance record (Table 1) is laid out for use by the tester. Not all operations need be performed for every test, and space has been provided for recording last service dates for motors, for instance, thus insuring that no item is overlooked. This list, when completed and signed by the tester, should be delivered to the test engineer for inspection. The engineer will be apprised by the list of the condition of the test stand at the end of the last test, and it may contain information which will influence his analysis of former data. The list will also inform the engineer of the condition of the stand prior to the current test and enable him to make necessary alterations or adjustments before the test starts.

The turbosupercharger should be accompanied by written instructions from the engineer for installing it. Test crews having even moderate experience do not need detailed instructions for a standard installation, but explicit instructions accompanied by diagrams should be given concerning special equipment.

The engineer should be notified when the unit installation is complete. His subsequent activities are governed by his confidence in the experience, interest, knowledge, preciseness, judgment, and reliability of the tester on the stand, and on the type of test to be run. If the test is to be of parts endurance, the engineer may do no more than check the mounting bolts and oil line connections, or if he is fortunate enough to have an experienced and reliable tester on the stand, all of the checks described below may be performed by the tester himself. Upon his arrival on the stand, the engineer first ascertains that the unit has been correctly aligned and installed in accordance with instructions. He checks the tightness of the mounting bolts, oil line connections, and tachometer cable. He visually inspects manometer lines for sharp bends or U-shaped dips in which moisture can collect. He sees to it that all lines are securely lagged.

TABLE 1

TEST STAND MAINTENANCE RECORD

Each item is to be initialed after it has been checked.

Unit No. — Stand No.
Type of Test — Date

Job	Comments	Initial
Test cell	Clean and oil-free	
Air filter	Cleaned	
Stand lights	Cleaned, bulbs replaced	
Hydraulic platform	Clean, lines pressure-checked	
Hydraulic lift	Chain link wear, rollers and track cleaned and oiled	
Steam lines	Joints cleaned and oiled, leak-free	
Ductwork	Clean, gaskets changed, insulation intact, correct for next setup	
Furnace	Clean, breaks repaired in liner or shell	
Fuel tank	Filter cleaned, fuel analysed for impurities	
Fuel lines	Cleaned, pressure-checked	
Fuel pump	Date of last service, gaskets, holds pressure	
Fuel nozzle	Cleaned, force of spray	
Oil system	Lines and tank flushed, lines pressure-checked, tank refilled, oil free from solvent	
Dryer	Clean, reactivated, cooled	
Intercooler	Cleaned, water lines cleaned and pressure-checked	
Expansion turbine	Clean, bearings worn	
Adjustable orifices	Clean, date of last calibration	
Tachometer	Cable clean, free from kinks, instrument checked	
Blowers	Clean, oiled	
Motors	Greased, date of last service, condition of wiring	
Valves	Proper seating, condition of packing	
Controls	Motors cleaned and oiled, date of last service, condition of wiring. Ease of operation, full travel, unused valves locked	
Pressure gauges	Calibrated, reset if necessary, lines cleaned and pressure-checked	
Thermocouples	Last calibrated, wire checked for breaks or shorts, potentiometer calibrated, battery renewed	
Manometers	Cleaned, refilled, leveled. Lines clean, blown out, pressure-checked	
Mounting bolts and nuts	Condition of shank heads and threads—replaced	
Clamps	Warped, out-of-round, lips sprung	

Tester ....

Test Observer ....

Initialled by Test Engineer ....

Correct testing procedure can be indicated by assuming that an altitude test of the entire turbosupercharger is being made. Valves (77), (78), and (79) are open (Fig. 7), valves (80), (84), and (86) are closed, and connection (*A*) is broken, venting the furnace inlet to atmosphere (Table 2). The engineer introduces compressed air into the compressor ductwork

TABLE 2

VALVE SETTINGS

Type of Test	Valves Open	Valves Closed	Connections Broken	Controls Operative
Steam pressure..	84	77, 78, 79, 80, 81, 82, 83, 85, 86	B, C, E	1, 2, 3, 26, 28
Regenerative...	80, 81, 86	77, 78, 79 (82, 83)[a] 84, 85	B, D	33 2, 3, 26, 41
Ejector........	80, 86	77, 78, 79, 81, 82 83, 84, 85	A, B, C, D	1, 2, 3, 48, 73
Altitude....... (with furnace)	79, 81, 86	(77, 78)[a] 80, 82, 83, 84, (85)[a]	A, D.	1, 2, 3, 33, 41, 48, 61, 62, 73

[a] ( ) Indicates valves open for prestart work; they are closed upon starting.

system through valve (85) until the pressure inside the ducts is about 10 psi (gauge). If the ducts will not hold this pressure with less than 0.2 psi pressure drop in five minutes, all joints and connections are checked with soapy water until the leaks are found and repaired. When the ducts are leak-free, the compressed air is allowed to escape from them; then valve (85) is closed and the dryer started. The air is circulated through the ducts and the dryer by the auxiliary dryer blower. While the dryer is operating, the engineer will check in the control room to be sure that log-sheet headings have been properly filled in, potentiometers balanced, manometers and inclinometers leveled and the scales set to correspond with the local barometric reading, the oil scale filled, and the adjustable orifices set for starting. As the dew point inside the ducts approaches the desired level, the oil should be preheated for starting with control (2). The desired dew point being reached, the dryer and blower are turned off; valves (77), (78), (82), and (83) are closed; valves (81) and (86) opened; and the cell blower (47)

is turned on. Controls (3), (26), (61), (62), (67), and (73) are set full-open; controls (1), (41), and (48) are full-closed; and the unit is ready to start.

Testing

While the observer puts pressure on the hand oil pump to insure adequate lubrication of the turbosupercharger for starting and the engineer watches all instruments for possible indications of trouble, the tester slowly opens control (1) until manometer (50) registers a considerable negative pressure. He then opens control (48) until the turbosupercharger is rotating at idling speed. As the main oil pressure (5) reaches the low limit, the observer shuts off the hand pump and returns to record the start on the test log. With the turbosupercharger idling, the tester turns on the furnace ignition (36) and the fuel pump (39). He then slowly opens control (41) until the furnace ignites. Furnace ignition (36) is then turned off, and the unit is operating and ready to set the first point. Controls (67) and (26) are not used for this type of testing, and remain open throughout the period of operation. Control locks should be provided so that any control not in use may be locked open or shut, as required. Controls (61) and (62) control the pressure and temperature drop across the expansion turbine, thereby controlling compressor-inlet conditions; controls (1) and (48) control the air flow through the turbosupercharger turbine, and control (41) regulates the temperature of the hot gases to the turbine. This permits great flexibility of testing, in that a wide range of air flows and pressure ratios is obtainable for a given turbine speed.

When a point stipulated by the engineer has been set and he is satisfied that the unit is stable, a reading is taken as quickly as is compatible with accuracy. Table 3 is a fictitious, sample test log kept by the observer. The tester, who is responsible for the operation of the unit and the test equipment, has few readings to take, and none that requires calculation. The engineer takes readings requiring specialized knowledge, makes most calculations, and plots the points on the curve.

TABLE 3

Test Log

Test Log No.— 11   Model No.— XC-1000

Unit Serial No.— 102   Manufacturer — Ellis Corp.

General Data / Cell

Time	a.m. or p.m.	Run-ning Time		Tach. rpm	Comp Air Flow #/Min	Tur-bine Air Flow #/Min	Vibra-tion Kc/Sec	True Wet Barom-eter "Hg	Dew Point °F	Endur-ance Time	Cell Temp °F	Cell Press "Hg A	Steam Press psi	Ejector Press "Hg A	Fuel Press psi	Fuel Flow #/Min
8 00	p m															
8 45	"	0 00	Start	2,000				30 01	−68	0 00						
9 15	"			10,000	167	1100	102	"	"		72	29 98	150	19 45	60	9 90
9 23	"			12,000	200	558	384	"	"		71	"	152	15 82	61	10 04
9 35	"			14,000	233	542	422	"	−67		70	"	153	17 98	60	10 52
9 45	"			16,000	206	550	403	"	"		68	29 87	157	17 55	63	11 10
10 00	"			18,000	300	357	390	30 00	"		67	"	160	17 27	64	11 43
10 15	"			20,000	333	383	398	"	"		68	"	165	16 89	65	12 25
10 30	"	1 45		22,000	367	585	411	"	−66	0 00	68	"	168	16 25	66	12 78
11 05	"			6,050	417	240	104	30 00	−65		67	29 87	145	26 04	62	11 08
11 25	"			12,400	351	240	116	"	"		67	"	151	24 62	62	19 92
11 50	"			15,000	290	240	117	"	−64		66	"	155	23 63	63	10 98
12 15	a m			20,300	212	240	109	"	"		65	"	160	17 54	63	12 54
12 45	"			21,300	167	240	108	29 98	−63		64	29 85	162	18 77	63	12 59
1 05	"			22,000	133	240	106	"	−62		64	"	162	17 46	64	11 01
1 20	"			23,200	116	200	107	"	−61		64	"	163	16 25	65	13 46
1 40	"			25,000	102	240	104	"	−60		64	"	166	11 32	65	14 67
2 00	"			30,950	83	240	110	"	−60		63	"	170	11 09	66	12 93
2 20	"	5 35		40,000	75	240	115	"	−58	0 00	63	"	190	5 47	66	19 98
3 05	"			22,790	218	185	212	29 96	−55		62	29 84	162	15 40	64	12 57
3 10	"			22,790	203	185	201	29 97	−50		61	29 85	162	17 82	65	12 03
3 30	"			22,790	198	184	189	"	−41		63	"	163	20 13	64	11 65
4 15	"			22,790	173	184	151	"	−36		63	"	162	22 22	64	11 17
4 40	"			22,700	156	184	158	29 98	−30		63	29 96	162	24 43	63	10 72
5 05	"			22,790	142	183	179	"	−8		64	"	162	26 62	64	10 18
5 10	"			22,790	128	182	207	"	+2		64	"	163	28 90	64	9 84
5 25	"	5 30		22,790						0 00						
5 40	"	8 55	Stop							0 00						

Air Brake / Expansion Turbine

Time	Air Brake rpm	Oil Press psi	Orifice Drop "H₂O	Temperature In °F	Temperature Out °F	Pressure In "H₂O	Pressure Out "Hg	Exp. Turbine Temperature In °R	Temperature Out °R	Pressure In "Hg A	Pressure Out "Hg A
8 00											
8 45	500	15									
9 15	900	28	11 43	72	75	29 97	30 35	532	458	30 15	24 72
9 23	1100	30	15 32	71	95	29 96	32 13	540	416	30 32	22 51
9 35	1300	34	19 21	70	115	29 94	34 38	531	373	30 34	20 00
9 45	1500	37	21 10	68	134	29 91	36 72	530	340	30 32	18 78
10 00	1700	40	26 09	67	154	29 88	38 95	529	325	30 33	17 42
10 15	1900	42	30 86	68	174	29 86	40 98	528	312	30 33	16 21
10 30	2100	44	33 02	68	195	29 83	43 12	529	300	30 34	14 84
11 05	1500	40	19 10	67	135	29 90	36 75	544	490	30 01	29 00
11 25	1500	40 5	19 12	67	135	29 90	36 75	540	487	30 02	27 00
11 50	1500	41	19 15	66	124	29 90	36 75	542	489	30 01	25 00
12 15	1500	41	19 11	65	113	29 90	36 75	545	490	30 01	22 00
12 45	1500	41	19 10	65	133	29 90	36 75	543	488	30 01	21 00
1 05	1500	41	19 15	64	132	29 90	36 75	544	489	30 00	20 00
1 20	1500	41	19 11	64	112	29 90	36 75	544	489	30 01	19 00
1 40	1500	41	19 12	63	131	29 90	36 75	543	488	30 01	17 00
2 00	1500	41	19 15	63	151	29 90	36 75	543	488	30 02	14 80
2 20	1500	41	19 16	63	171	29 90	36 75	544	489	30 01	11 00
3 05	900	30	11 03	62	66	29 98	30 12	590	490	32 10	29 10
3 10	1000	32	12 92	63	75	29 99	30 25	579	490	32 08	27 11
3 30	1100	34	13 81	63	86	29 99	31 08	581	490	32 11	25 10
4 15	1300	36	15 00	63	98	29 89	32 25	580	490	32 67	23 12
4 40	1300	36	16 19	64	110	29 90	33 67	590	491	33 92	21 11
5 05	1400	40	17 28	64	122	29 90	35 11	580	490	34 14	19 10
5 10	1500	41	18 37	64	133	29 90	36 43	580	490	34 67	17 10
5 25											
5 40											

Lubrication System / Speed

Time	Oil Press psi	Oil Temperature In °F	Oil Temperature Out °F	Rise °F	Oil Flow #/Min	Heat Rejec-tion Btu/Min	Oil Scale Lb	Oil Con-sumption #/Hr	N rpm	V ′/sec	0088/√T	u ′/sec
8 00												
8 45	25	100			0 5		115					
9 15	40	170	181	11	3 5	19	114 8		9 882	306	02363	606
9 23	47	172	184	12	5 0	30	114 7		11 821	610	02071	610
9 35	54	160	185	16	7 0	56	114 5		13 575	711	01763	711
9 45	63	168	187	19	8 5	84	114 3	0 7	15 499	812	01522	812
10 00	70	168	190	22	9 8	100	114 1		17 427	913	01235	913
10 15	76	168	193	25	11 0	126	114 0		19 168	1015	00976	1015
10 30	81	168	195	27	12 7	167	113 9		21 290	1117	00757	1117
								0 5				
11 05	41	172	177	5	1 2	3	111 7		5,860	306	0649	306
11 25	45	168	186	18	4 9	31	113 6		12,004	621	0898	621
11 50	70	165	188	23	9 2	109	112 5	0 3	18 001	914	0642	914
12 15	75	165	191	26	11 1	126	[illegible]		19 530	1135	0511	1135
12 45	77	165	192	28	11 9	167	113 3	0 1	20,515	1202	0465	1202
1 05	80	166	193	27	12 5	172	113 2		21,298	1320	0422	1320
1 20	85	165	191	28	13 1	169	[illegible]		22,516	1376	0357	1376
1 40	86	165	195	30	14 5	243	[illegible]	0 4	23 005	1715	0318	1715
2 00	88	164	200	36	18 6	302	112 9		29,943	1806	0256	1806
2 20	90	165	207	42	22 7	477	112 8		39,298	2080	01438	2080
								0 3				
3 05	85	167	195	28	11 5	189	112 6		22,000	1153	0456	1153
3 10	86	166	193	27	11 0	185	112 5		22 000	1153	0456	1153
3 30	85	165	192	27	[illegible]	187	112 4	0 3	22 000	1153	0456	1153
4 15	65	164	192	28	12 8	183	112 3		22,000	1153	0456	1153
4 40	85	163	191	28	13 1	186	112 2	0 1	22 000	1153	0456	1153
5 05	83	164	191	27	11 3	186	112 1		22,000	1153	0456	1153
5 10	84	165	192	27	13 5	188	112 0		22,000	1153	0456	1153
5 25									22 000	1153	0456	1153
5 40							111 9	0 4				

Fuel-oil type — Bunker "C"
Lube-oil type — Saybolt 130
Dustmark — No 2 No 10
Mount — 3-ft diam

Impeller diam — 12"
Number of blades — 27
Part No — 41370

Rotor diameter — 13.75"
Number of blades — 84
Part No — 25,880

TABLE 3 (Cont'd)

Complete Calibration of Turbosupercharger Unit

11-44— Date   Ejector and H G – Method of Drive
19— Test Stand No   19,352.70— Tachometer Constant

Turbosupercharger: Compressor System										Compressor Orifice								Turbine System								Turbine Orifice							Cooling Air Flow					
P_1	P_2	$\frac{P_2}{P_1}$	γ	T_1	T_2	ΔT	ΔT_i	$\frac{600 \Delta T_i}{V^2}$	$\frac{\Delta T_i}{\Delta T}$	Diameter	Inlet Press	Inlet Temp	Drop	W_c	Q	$\frac{Q}{N}$		P_3	P_4	$\frac{P_3}{P_4}$	T_3	H	C	$\frac{u}{C}$	T_4	Diameter	Inlet Press	Inlet Temp	Drop	W_t	Comp	E_t	Diameter	Inlet Press	Inlet Temp	Drop	Cooling Air Flow	Comments
"Hg A	"Hg A			°R	°R	°F	°F			In	"Hg A	°R	"H₂O	# Hr	CFM			"Hg A	"Hg A		°F	Btu/#	sec		°F	In	"Hg A	°R	"H₂O	# Hr	hp		In	"H₂O	°R	"H₂O	# Hr	
																																	Not installed per engineer					Engineer pressure-checked ducts. Leakage 1.2" H₂O in 5 minutes. Operated dryer for 0.25 hr
																																						Start – warm up to basic standard turbosupercharger calibration curve
34.70	30.45	1.231	00034	457	532	74.4	27.38	654	3.71	10	24.70	457	7.5	14,730	2925	240		29.99	20.00	1.50	980	36.2	1296	391	490	10	30.00	529	49.7	66,000	71	.076						
21.80	30.42	1.352	00086	416	530	115.1	36.72	631	349	10	22.53	455	3.9	16,610	2930	259		29.99	17.24	1.74	1045	53.0	1361	432	520	10	30.00	529	11.9	33,530	132	.199						
20.58	30.40	1.483	11734	376	531	115.5	42.31	683	3306	10	20.58	453	5.7	18,550	3470	271		29.98	18.72	1.60	1157	34.7	1456	473	551	10	29.98	529	10.8	32,500	208	.298						
18.70	30.44	1.624	14036	341	530	160.0	46.29	586	3004	10	18.76	450	6.8	20,460	3565	285		29.62	18.29	1.62	1210	34.1	1553	514	575	10	29.95	529	11.6	33,000	290	.198						
17.40	30.48	1.758	17163	328	529	223.4	54.98	437	3052	10	17.40	445	8.3	22,910	4640	297		29.37	17.48	1.68	1333	59.8	1641	545	647	10	29.93	526	12.2	33,490	185	.187						
16.19	30.35	1.887	19500	313	526	251.2	60.96	301	2990	10	16.19	441	10.5	24,370	5745	309		29.88	17.01	1.71	1402	62.9	1724	576	691	10	29.90	528	12.1	33,960	460	.556						
14.82	30.31	2.045	22441	304	529	228.0	67.00	3306	2965	10	14.92	436	12.6	26,830	6640	321		28.83	16.30	1.77	1505	71.5	1912	616	715	10	29.53	528	12.8	34,700	504	.599						
																																						End of standard calibration curve, begin u/c curve at constant P_3/P_4 = 3.00
29.62	30.00	1.05	01300	925	925	1.54	1.50	.985	.0974	10	29.65	847	32.9	25,010	900	167		29.95	9.98	3.00	1510	130	2450	125	704	9	29.98	5.5	18.1	14,400	23.00	.500						
27.28	30.12	1.14	05776	850	852	2.38	2.25	.966	.0976	10	27.31	800	23.7	21,812	940	155		29.95	9.96	3.00	1516	130.5	2451	199	706	9	29.96	525	18.0	14,395	24.10	.573						
25.26	30.16	1.23	08034	809	812	3.86	3.51	.967	.0978	9	25.27	761	38.8	17,006	800	143		29.95	9.98	3.00	1518	130.5	2451	306	706	9	29.96	525	18.0	14,395	24.62	.639						
22.92	30.21	1.26	09987	754	756	5.02	4.64	.968	.0980	9	22.95	701	27.3	13,010	830	131		29.95	9.98	3.00	1510	130	2450	360	704	9	29.98	525	18.0	14,392	25.43	.654						
21.18	30.25	1.47	11530	703	708	5.21	5.00	.969	.0982	8	21.21	655	30.2	9,998	750	119		29.95	9.98	3.00	1503	129.5	2449	410	702	9	29.98	523	18.0	14,395	26.08	.676						
20.70	30.30	1.51	12370	651	657	6.56	6.23	.970	.0984	8	20.79	603	23.9	8,002	650	108		29.94	9.97	3.00	1503	129	2448	455	701	9	29.97	523	18.0	14,392	27.79	.661						
19.53	30.33	1.60	14226	602	608	6.62	6.35	.971	.0985	8	19.55	563	18.1	7,009	550	.097		29.94	9.97	3.00	1510	130	2450	500	704	9	29.97	523	18.0	14,390	26.32	.676						
17.20	30.45	1.82	18408	558	564	6.74	6.48	.972	.0986	7	17.33	545	28.6	5,992	451	.086		29.94	9.97	3.00	1503	129.5	2449	563	702	9	29.97	523	18.0	14,390	29.21	.637						
15.02	30.48	2.10	23304	510	517	6.90	6.72	.974	.0987	7	15.04	510	22.4	5,010	350	.075		29.94	9.97	3.00	1500	128.3	2447	641	701	9	29.97	523	18.0	14,387	29.73	.586						
11.14	30.82	2.82	34007	406	503	7.06	6.67	.975	.0988	7	11.18	498	15.5	4,500	252	.0641		29.94	9.97	3.00	1500	128.5	2447	842	701	9	29.97	523	17.9	14,385	30.30	.411						
																																						End of u/c curve, begin compressor calibration curve at constant 22,000 rpm
30.30	32.53	1.11	02807	1470	1700	232	44.1	.190	.201	5	29.25	847	4.73	1,308	528	240		29.94	15.43	1.94	1642	86	2138	475	852	5	29.93	523	20.7	11,100	174	.062						
27.70	32.83	1.23	0663	1354	1570	221	63.2	.438	.452	5	27.75	800	4.44	1,217	502	231		29.93	17.91	1.64	1645	73	2233	501	853	5	29.94	524	20.7	11,100	156	.114						
25.30	33.10	1.47	1042	1188	1402	214	82.4	.702	.670	5	25.35	790	4.53	1,126	476	212		29.93	20.16	1.65	1643	60	2037	622	852	5	29.94	524	20.8	11,130	127	.147						
23.80	33.75	1.64	1374	1057	1260	206	100.6	.801	.673	5	23.85	782	4.42	1,035	430	183		29.94	22.24	1.45	1647	48	1834	687	854	5	29.95	525	20.8	10,990	101	.190						
21.40	34.22	1.65	1685	911	1000	180	121.8	.942	.687	5	21.43	683	4.31	944	375	160		29.94	24.45	1.28	1649	35	1616	742	855	5	29.94	525	20.8	11,110	82	.205						
19.90	34.83	1.98	1921	821	991	172	136.8	.951	.693	5	19.95	651	4.20	853	325	147		29.94	26.70	1.15	1651	22	1417	903	856	5	29.95	525	20.7	11,100	48	.239						
17.50	35.30	2.02	2302	700	860	163	153.4	.940	.700	5	17.55	649	4.00	700	295	132		29.95	28.93	1.04	1648	8.5	1235	860	855	5	29.96	525	20.7	11,100	22	.293						
										5	15.35	822	3.96	644	273	124																						Surge point
																																						Dew point rising rapidly. Stop to dry out air in ducts

Nozzle ring diameter	– 12"
Number of vanes	– 25
Part No	–34 384

Special Parts on Test –
None

Shift	1st	2nd	3rd
Tester	Smith	Black	Anderson
Test Observer	Green	Jones	White
Test Engineer	Johnson	Kelly	Brown

TABLE 4

TURBOSUPERCHARGER CALCULATIONS—TEST-STAND METHOD

The following data are from line 9 of Table 19:

P_1—compressor inlet pressure	14.82	"HgA
P_2—compressor outlet pressure	30.31	"HgA
P_3—turbine inlet pressure	28.83	"HgA
P_4—turbine outlet pressure	16.30	"HgA
T_1—compressor inlet temperature	301	°R
T_2—compressor outlet temperature	529	°R
T_3—turbine inlet temperature	1505	°F
Rpm—indicated turbine speed	22,000	
Compressor orifice diam	10"	
Compressor-orifice inlet pressure	14.82	"HgA
Compressor-orifice inlet temperature	436	°R
Orifice constant (smooth approach)	0.99	
Pressure drop across orifice	12.6	"H_2O
Turbine orifice diam	10"	
Turbine-orifice inlet pressure	29.83	"HgA
Turbine-orifice inlet temperature	528	°R
Orifice constant (smooth approach)	0.99	
Pressure drop across orifice	12.8	"H_2O
Impeller diam	12"	
Rotor diam at center line of nozzle ring	12"	
Tachometer gearing constant	$\frac{19.352}{20}$	

CALCULATIONS

$$\text{Compressor pressure ratio} = \frac{P_2}{P_1} = \frac{30.31}{14.82} = 2.045$$

$$Y = \frac{(P_2)}{(P_1)}^{0.283} - 1 = 0.22441$$

$$YT_1 = \text{adiabatic temp rise across compressor}$$
$$= (0.22441)\ (301) = 67.60$$

$$\Delta T = \text{indicated temp rise across compressor} = T_2 - T_1$$
$$= 529 - 301 = 228$$

$$\frac{YT_1}{\Delta T} = \text{temp rise ratio} = \frac{67.60}{228} = 0.2965$$

$$N = \text{actual turbine speed} = \text{rpm} \times \text{tachometer const}$$
$$= \frac{19.352}{20}\ (22{,}000) = 21{,}290$$

$$V = \text{impeller tip speed} = \text{impeller circum}^* \times \text{rps}$$
$$= \frac{12}{12}\,\pi\,\frac{(21{,}290)}{(60)} = 1117\ \text{fps}$$

$$\frac{6088\ YT_1}{V^2} = \text{pressure coef} = \frac{6088\ (67.60)}{(1117)^2} = 0.3308$$

* In feet

TABLE 4 (*Cont.*)

TURBOSUPERCHARGER CALCULATIONS—TEST-STAND METHOD

W_c = compressor air flow

$$= 413.5 \text{ (orif diam)}^2 \text{ (orif const)} (\sqrt{\text{press. drop}}) \left(\sqrt{\frac{\text{inlet press.}}{\text{inlet temp}}}\right)$$

$$= 413.5\ (10)^2\ (0.99)\ (\sqrt{12.6})\left(\sqrt{\frac{14.82}{436}}\right) = 26{,}820 \text{ lb per hr}$$

Turbine air flow (W_t), cooling air flow (CAF), and air-brake air flow (ABF) calculated as above.

$$Q = \text{vol flow of air} = 0.01257\ W_c \frac{T_1}{P_1}$$

$$= 0.01257\ (26{,}820) \left(\frac{301}{14.82}\right) = 6840\ CFM$$

$$Q/N = \text{load coef} = \frac{6840}{21{,}290} = 0.321$$

$$\text{hp} = \text{power absorbed by compressor} = \frac{778 W_c\ (\Delta T) \text{ sp ht of air}}{33{,}000}$$

$$= \frac{778\ (26{,}820)\ (228)\ (0.243)}{33{,}000} = 584 \text{ hp}$$

$$\frac{P_3}{P_4} = \text{turbine pressure ratio} = \frac{28.83}{16.30} = 1.77$$

H = available energy content of driving gas. Knowing $\frac{P_3}{P_4}$ and T_3, H may be read from graph (NACA-ARR No. 4B25) = 71.5 Btu per lb

C = gas velocity at nozzle discharge = 223.7 (nozzle const) ($\sqrt{H}$) = 223.7 (0.96)($\sqrt{130.5}$) = 1812 fps

u = bucket velocity = circum of rotor at center line of nozzle ring* × rps

$$= \frac{12}{12}\pi\left(\frac{21{,}290}{60}\right) = 1117 \text{ fps}$$

$$\frac{u}{C} = \text{velocity ratio} = 0.616$$

W_t = turbine air flow = 34,780 (see W_c)

$$E = \text{turbosupercharger efficiency} = \frac{W_c}{W_t} \text{(sp ht of air)} \frac{\Delta T}{H}$$

$$= \frac{26{,}820}{34{,}780} (0.243) \frac{228}{71.5} = 0.599$$

*In feet

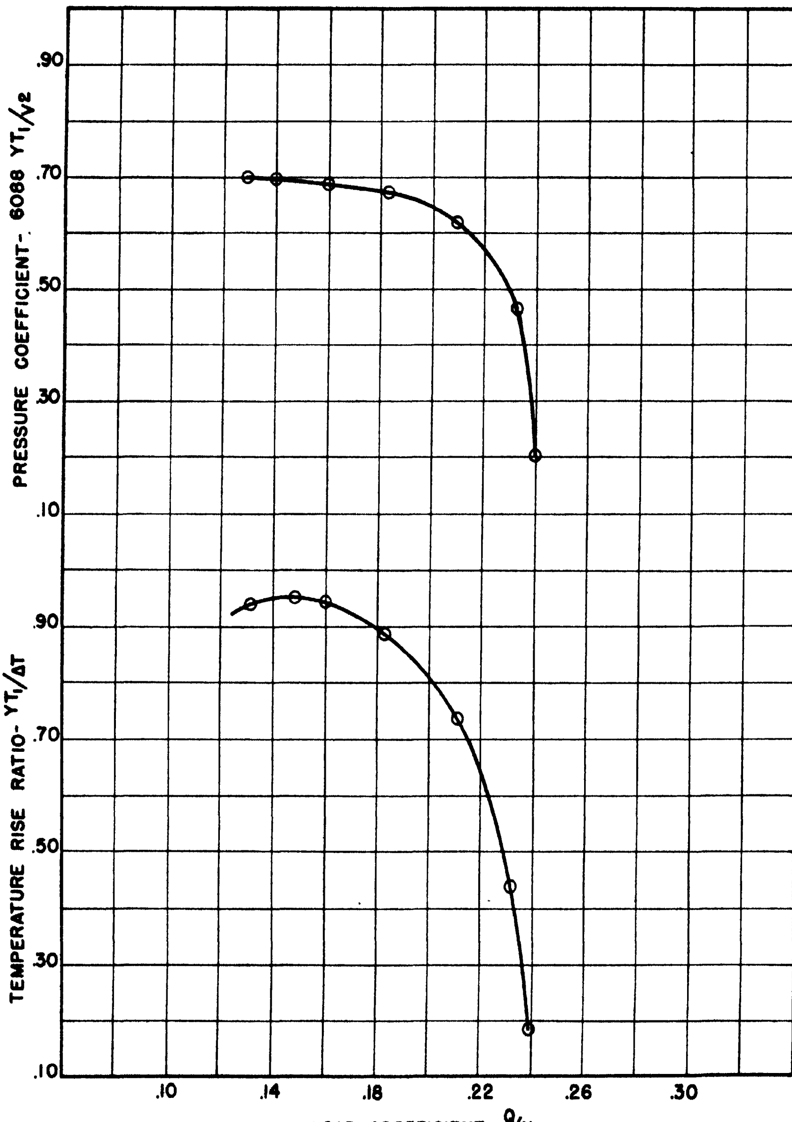


Fig. 8. Load Coefficient vs. Temperature Rise Ratio and Pressure Coefficient.

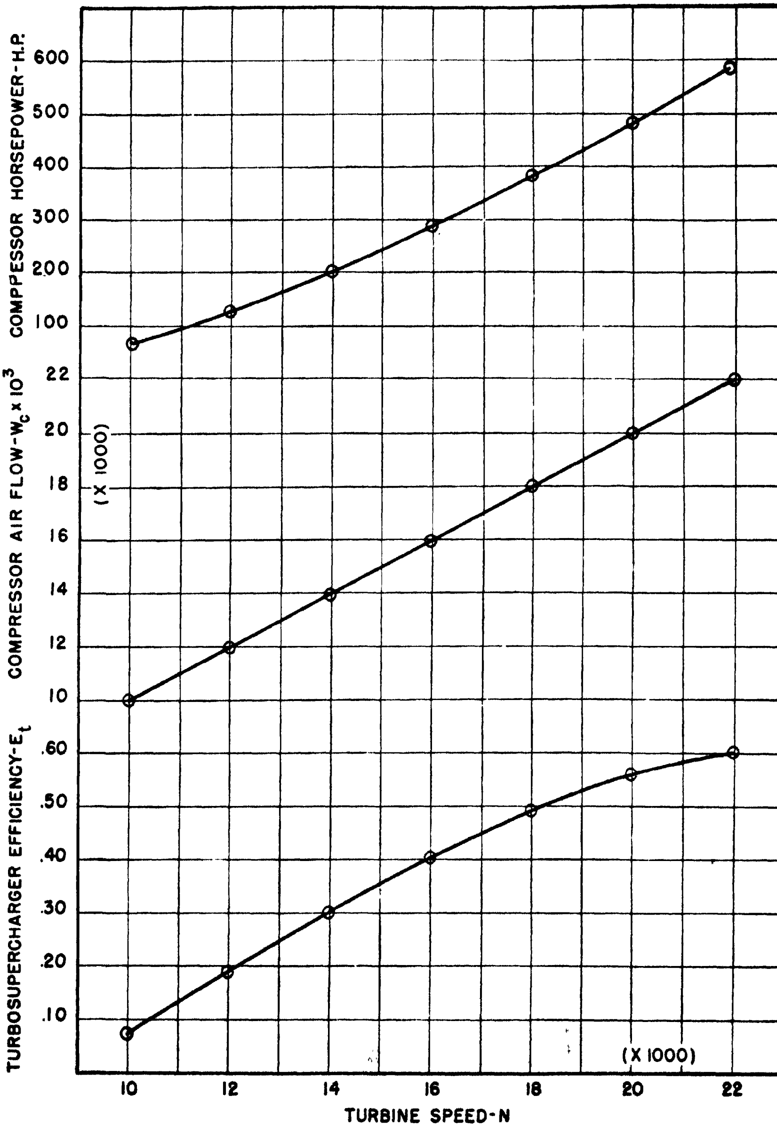


Fig. 9. Turbine Speed vs. Efficiency, Compressor Air Flow, and Horsepower.

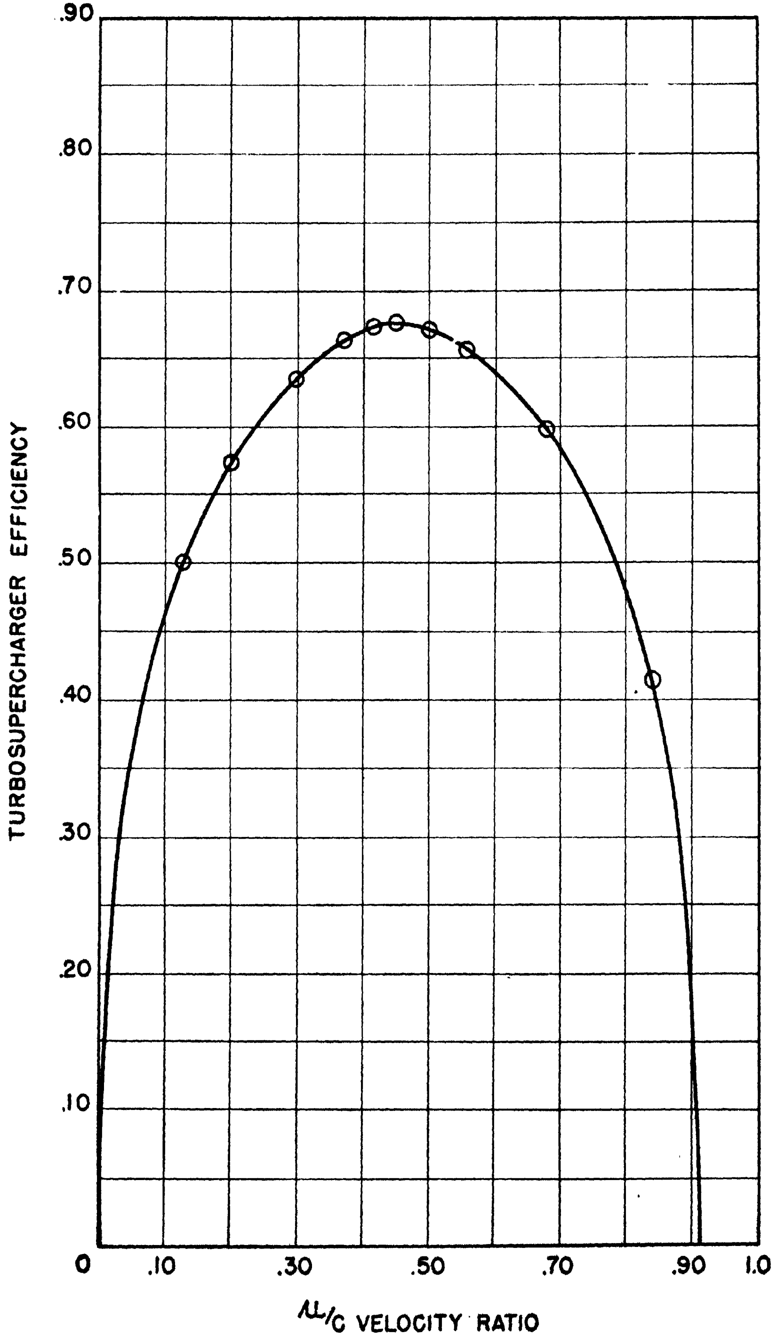


Fig. 10. Velocity Ratio vs. Efficiency.

The test observer takes the remainder of the readings, makes simple calculations, and records on the test log the tester's readings and the engineer's readings and calculations. (See sample test log Table 3, calculation sheet in Table 4, and curves of Figures 8, 9, and 10.)

The engineer plots points on the curve as the turbosupercharger is run. If a point does not fall on the curve, that point and possibly part of the stand instrumentation may be checked. Everything that occurs after the engineer arrives on the stand is recorded on the test log. This running commentary, coupled with the maintenance record filled out by the tester prior to the test, gives a complete picture of all factors affecting analysis of test data.

PART III

AIRCRAFT GAS TURBINE EQUIPMENT

G. Geoffrey Smith, Foremost British Author of Gas Turbine Literature, Directing Editor of *Flight* Magazine and Author of *Gas Turbines and Jet Propulsion for Aircraft*, Comparing Notes on a Rolls-Royce Derwent Gas Turbine Unit with E. W. Hives, Managing Director of Rolls-Royce, Ltd.

Section A—American

Chapter XIII

Westinghouse Electric Corporation Jet Engines[1]

The Westinghouse Electric Corporation, South Philadelphia Plant, has developed the baby jet unit, which is known as the 9.5A model, Figure 1, and a large jet unit, the 19B model, Figure 2. In addition to these units already produced a propeller-drive gas turbine is being developed.

The propeller-drive gas turbine will be used, perhaps more extensively than the jet engine, when high power combined with efficient operation is required in planes operating up to 550 mph. For several years to come, the propeller-drive gas turbine will be favored in long-range heavy bombers and in long-range ships.

The 19B engine is good for 1400 hp at modern plane speeds. A conventional aircraft power plant of the same horsepower has more than twice the diameter of this jet engine. Engineers of the Westinghouse Electric Corporation believe they are making the smallest and lightest jet engine (for a given horsepower) in the world. This Westinghouse unit weighs only one half as much as the corresponding piston engine.

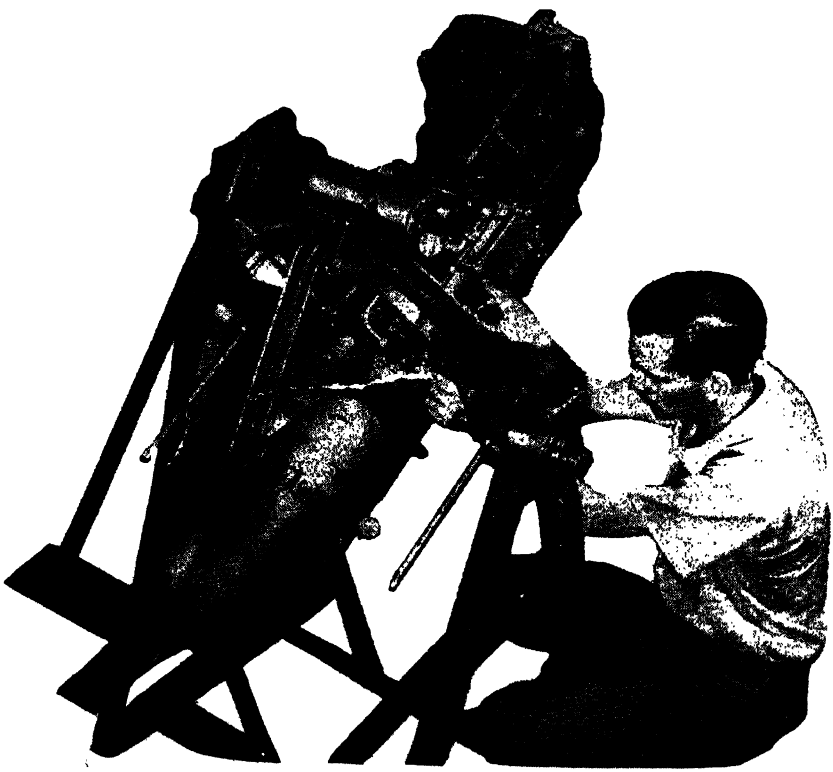
In contrast to the rocket, which carries its own oxygen for combustion, the jet engine brings in its air from the outside. Its basic operating principles are:

1. Air is pumped in by a compressor.
2. The air is heated by burning liquid fuel in the combustor.

[1] Abstract from *Mechanical Engineering.*

3. A portion of the energy of the hot combustion products, which have been expanded to several times their original volume, is used to drive a turbine, the sole purpose of which is to supply power to keep the compressor going.

4. The remaining horsepower is not delivered to a shaft, but appears in the form of a high-velocity jet. It is reaction of this jet that propels the aircraft.



Courtesy of Westinghouse Electric Corporation

Fig. 1. The 9.5A Model Jet Unit.

It is possible to take more power out of the turbine than is needed to drive the compressor, and this surplus power can then be used to drive a propeller. Such an arrangement, in which only a small part (20 per cent) of the energy remains in the jet, is known as a *gas turbine propeller drive.*

In contrast to the present-day piston engine, the jet engine

has, in reality, only one moving element, the compressor and the turbine in line on one shaft. This is one reason for the streamlined appearance of these jet engines.

Another reason for the small diameter is that Westing-

Courtesy of Westinghouse Electric Corporation

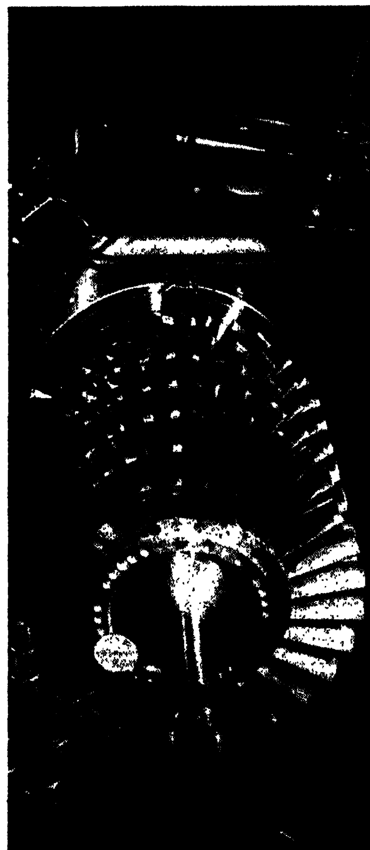
Fig. 2. The 19B Jet-propulsion Engine.

house has selected the axial-flow type of compressor as in Figure 5. In contrast to the centrifugal compressor, which utilizes centrifugal force to pump up the medium and requires

Courtesy of Westinghouse Electric Corporation

Fig. 3. The Turbine wheel and Shaft to the Compressor of the 19B Jet Unit.

large diameter, the axial-flow compressor is like a fan with many blades that pushes the air backward toward the combustion chamber. In this six-stage compressor, the rotating blades go around at a speed of 18,000 rpm—that is, 300 times per second. At 18,000 rpm the compressor delivers air at the rate of 50 tons per hour to the combustion chamber.



Courtesy of Westinghouse Electric Corporation

Fig. 4. The Six-stage Compressor of the 19B Unit.

The combustion chamber resembles a perforated wastepaper basket, and the compressed air enters the burner baskets through these perforations. Fuel is sprayed in through a row of atomizing spray nozzles. A spark is used for ignition, but as soon as the flame has started, the ignition can be cut off because the combustion is continuous. The air particles spend only 1/100 sec in the combustion chamber. The rate of combustion is so intensive that in a given space 1000 times as much heat is released as in a conventional power-plant boiler. Brought to a temperature of 1500° F, the combustion products then enter the turbine, where they give off a good deal of their energy to drive the compressor. The tips of the turbine blades move at 800 mph, revolving so rapidly that the centrifugal pull on each turbine blade is 50,000 times its own weight.

As the air enters the engine, it first cools the lubricating oil. The aluminum oil cooler is located where it is subjected

to cooling air regardless of whether the airplane is flying or is on the ground.

After having gone through the turbine, the gases then enter the exhaust nozzle. From there, the jet exhausts as a 1200-mph gale. This exhaust nozzle is one in which the area, and thereby the velocity, of the jet can be varied by a movable tailpiece.

The accessory drive comprises accessories that serve the engine proper, and consist of:

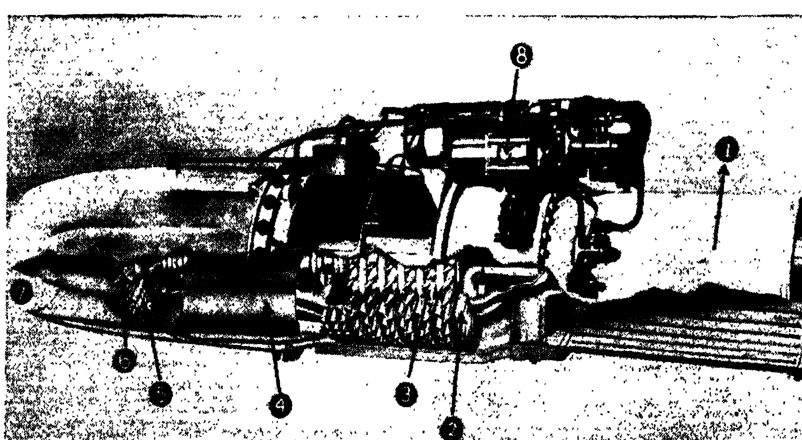
An *electric starter* to bring the engine up to the speed at which it can maintain itself.

A *fuel pump* to deliver fuel to the combustion chamber.

An *oil pump* to circulate the oil to the bearings and to the oil cooler, which is mounted in front of the engine where air cooling is available at all times.

An *overspeed control* to prevent the engine from running away.

An *electric tachometer* to give a visual indication of rpm to the pilot.



Courtesy of Westinghouse Electric Corporation

1—Inlet Duct
2—Compressor Stator Blades
3—Axial-Flow Compressor
4—Combustor
5—Turbine Stator Blades
6—Turbine Rotor
7—Jet Nozzle
8—Accessory Section

Fig. 5. Cutaway Drawing Showing the Major Parts of the 19B Unit.

The accessories that serve the airplane are:

A *generator* to provide electric current.

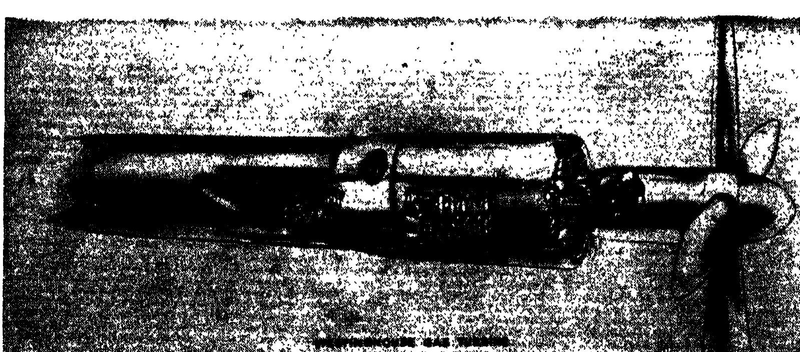
A *hydraulic pump* to furnish high-pressure oil for serving wing flaps, landing gear, and so forth.

A *vacuum pump* to operate the aircraft instruments.

The 9.5A engine has many of the same features. Its top speed is 34,000 rpm, 567 rev per sec. It was originally designed to power an American buzzless bomb, but shows some promise for driving small planes and, in a modified form, as a small mechanical-drive turbine to drive helicopters, cabin superchargers, and electric generators.

The engines built by Westinghouse thus far are all pure jet engines. However, the jet engine is essentially a power plant suited for high-speed aircraft. Such aircraft should have a clean, streamlined design and its engine should have a small area. Analysis of gas turbine propeller drives has proved that Westinghouse can build a gas turbine of only one half the diameter and between one half and three quarters of the installed weight of the present piston engine of the same power.

Studies show that on a large plane the gas turbine would give performance superior to that given by the piston engine in every respect, namely, in range of the aircraft, maximum-speed rate of climb, pay load, and take-off distance. In addition, the inherent simplicity of the gas turbine, the fact that



Courtesy of Westinghouse Electric Corporation

Fig. 6. Westinghouse Gas Turbine Aircraft Propeller Drive.

practically no cooling is needed, and that vibration and noise are much reduced as compared with the piston engine—all these factors will make the gas turbine propeller drive extremely attractive for future aviation (Fig. 6).

Already there is a demand for engines of 6000 and 10,000 hp. The Germans built an 8000-hp jet engine. A piston engine for such powers would be difficult to build, and it is likely that its size and complexity would result in a prohibitive cost. However, there appears to be no good reason why jet engines or gas turbines of 6000 or 10,000 hp cannot be built today.

Chapter XIV

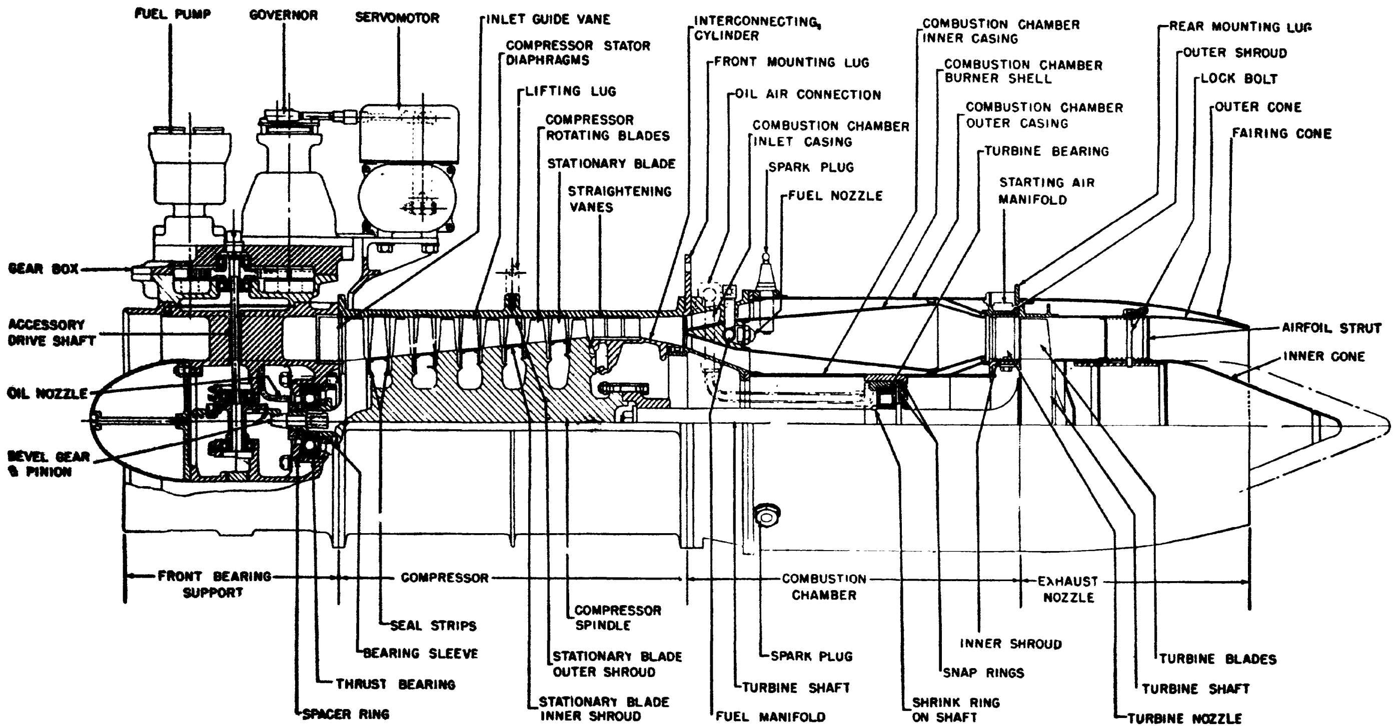
Service Instructions for Westinghouse Model X9.5A Jet Unit

The Westinghouse Model X9.5A jet propulsion engine is an axial-flow gas turbine power plant designed specifically for airborne jet propulsion. It is an experimental engine only, and is intended for use as a main power plant, although it may be used as a booster for conventional, reciprocating engine-propeller-driven aircraft.

The engine, shown in Figure 1, is composed of a series of approximately cylindrical sections that are axially aligned. Cross-sectional diameters of the sections vary from $7\frac{7}{16}$ to $3\frac{1}{2}$ in. Starting from the air-inlet end, the engine contains essentially four main assemblies: a front bearing-support assembly (which forms the air-inlet duct and houses the accessory drive), an axial-flow compressor assembly, a combustion-chamber and turbine assembly, and an exhaust-nozzle assembly.

The general specifications for the engine are as follows:

Total dry weight (including accessories necessary for operation)	145 lb
Length over-all (tail-cone assembly retracted)	$49\frac{31}{32}''$
Length over-all (tail-cone assembly extended)	$52\frac{15}{32}''$
Diam at combustion chamber	$9\frac{1}{2}''$
Type of bearings	Ball and roller
Moment of inertia of rotating parts	0.226 in-lb per sec²
Military rating:	
Max speed	54,000 rpm
Max turbine inlet temperature	1500° F
Normal rating:	
Speed (approx)	28,000 rpm
Turbine inlet temperature	1200° F



Courtesy of Westinghouse Electric Corporation

Fig. 1. Longitudinal Section of Model X9.5A Jet Propulsion Engine.

The turbine-driven accessories needed for operation are a vacuum pump, a fuel pump, and an all-speed governor. These accessories, mounted on a gear box which is located on top of the front-bearing support assembly, are driven by reduction gearing housed in the front-bearing support assembly and in the gear box. Also included as standard accessories are an oil reservoir, servo motor, tachometer generator, relief valve, fuel shut-off valve, check valve, fuel strainer, booster coils, spark plugs, and thermocouples.

A throttle control and two switches govern engine operation. The throttle control energizes the servo motor which, in turn, sets the governor. A switch controls the electrically operated fuel shut-off valve. Another switch closes the ignition circuit to ignite the fuel mixture.

Unlike the conventional reciprocating gasoline engine, once the fuel mixture is fired, combustion is self-sustaining in the jet engine. In addition, the jet engine requires no warm-up period when starting.

The following definitions will be used:

The *front* of the engine refers to the air-inlet end.

The *right* and *left* refer to the sides of the engine as viewed from the rear.

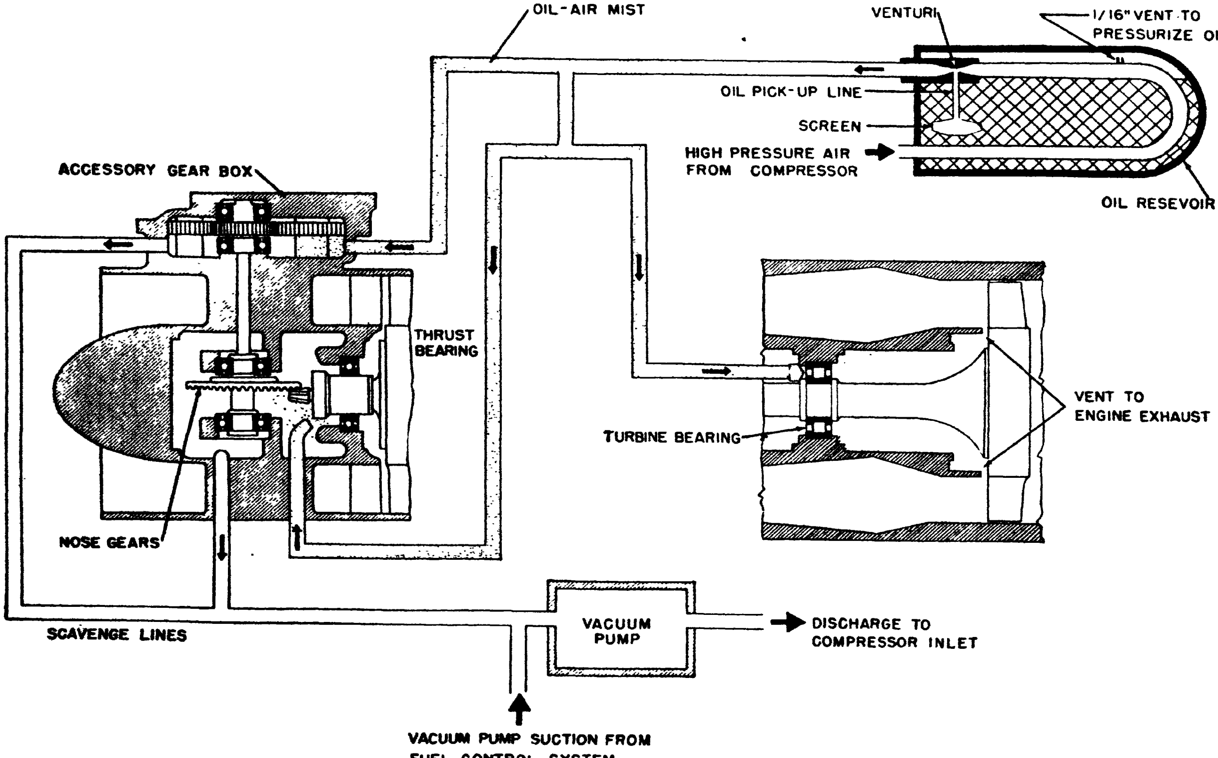
The terms *clockwise* and *counterclockwise* refer to the direction of rotation of revolving parts as viewed from the rear of the engine, or in the case of accessory drives, as shown by a plan view of the accessory gear box.

The *main coupling* is the coupling joining the turbine shaft to the compressor spindle.

Lubrication System

The lubrication system is designed to supply an oil-air mist to gears and bearings in sufficient quantities to lubricate these parts and to dissipate the heat from them. Used oil is not recirculated, but is scavenged and burned. Lubricating oil must be in accordance with Army-Navy specification AN-0-6. A schematic diagram of the lubricating system is shown in Figure 2.

To generate the oil-air mist, high-pressure air is bled from the rear of the compressor. The air is first piped through an



Courtesy of Westinghouse Electric Corporation

Fig. 2. Schematic Drawing of Lubrication System.

oil reservoir where heat from the compressed air warms the oil to prevent it from congealing at the low temperatures of high altitudes. The air, which is vented to the reservoir to pressurize the oil, is then led to a venturi. Oil is driven up a tube connected to the venturi throat by the difference in pressure between the oil reservoir and the venturi throat. Oil and air are mixed in the venturi and carried through feeder piping to the following lubrication points: the gears in the gear box, the spiral gear and pinion, the compressor spindle bearing in the front-bearing support assembly, and the turbine bearing in the combustion-chamber assembly. Resistance of the feeder piping controls the relative amount of oil-air mist let to each point. The oil-air mist is blasted into the bearings at a velocity of 1000 ft per sec.

Oil accumulating in the front-bearing support assembly and the gear-box housing is scavenged by the vacuum pump and discharged into the air-intake ducts of the bearing-support assembly. The discharged oil is carried through the compressor into the combustion chamber by air drawn through the engine.

Oil sprayed on the turbine bearing is carried along the turbine shaft and out into the engine exhaust through the space between the turbine nozzle and rotating blades. The air-pressure drop between the bearing and turbine nozzle provides the scavenging force.

Lubricating-system Components

Vacuum Pump

The vacuum pump is located on top of the forward, left-hand area of the upper gear-box housing (Figs. 3 and 4). Refer to the *Accessories* paragraph, page 268, for a general description of the pump.

Oil Reservoir

In the standard arrangement of parts, the oil reservoir is not mounted on the engine, but is located horizontally on the air frame above the compressor cover and just forward of the front mounting lug (Fig. 5). The reservoir is cylindrical; its capacity is two quarts.

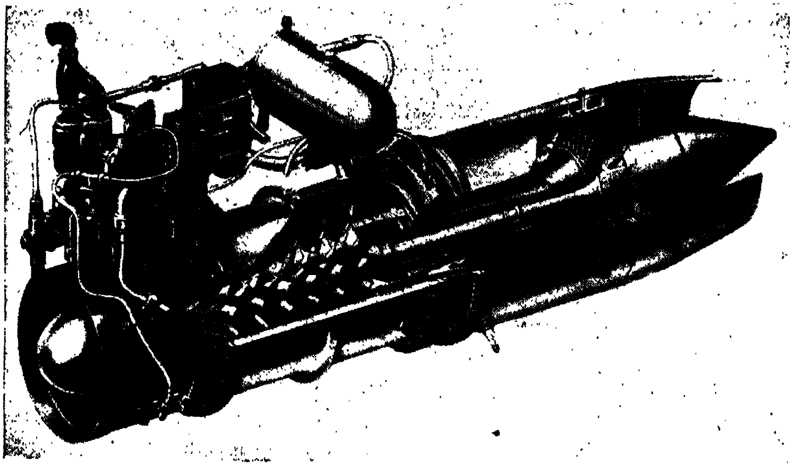


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Fig. 3. Cutaway View of Unit, Left Side, Showing Accessories.

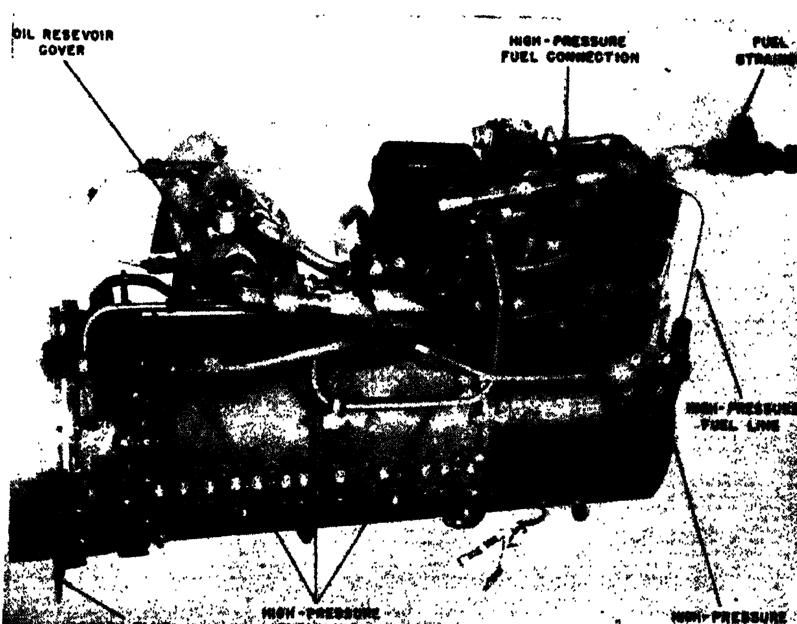

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Fig. 4. Right Side of Engine—Forward Sections, Accessories and Tubing Giving Identification of Inspection Parts.

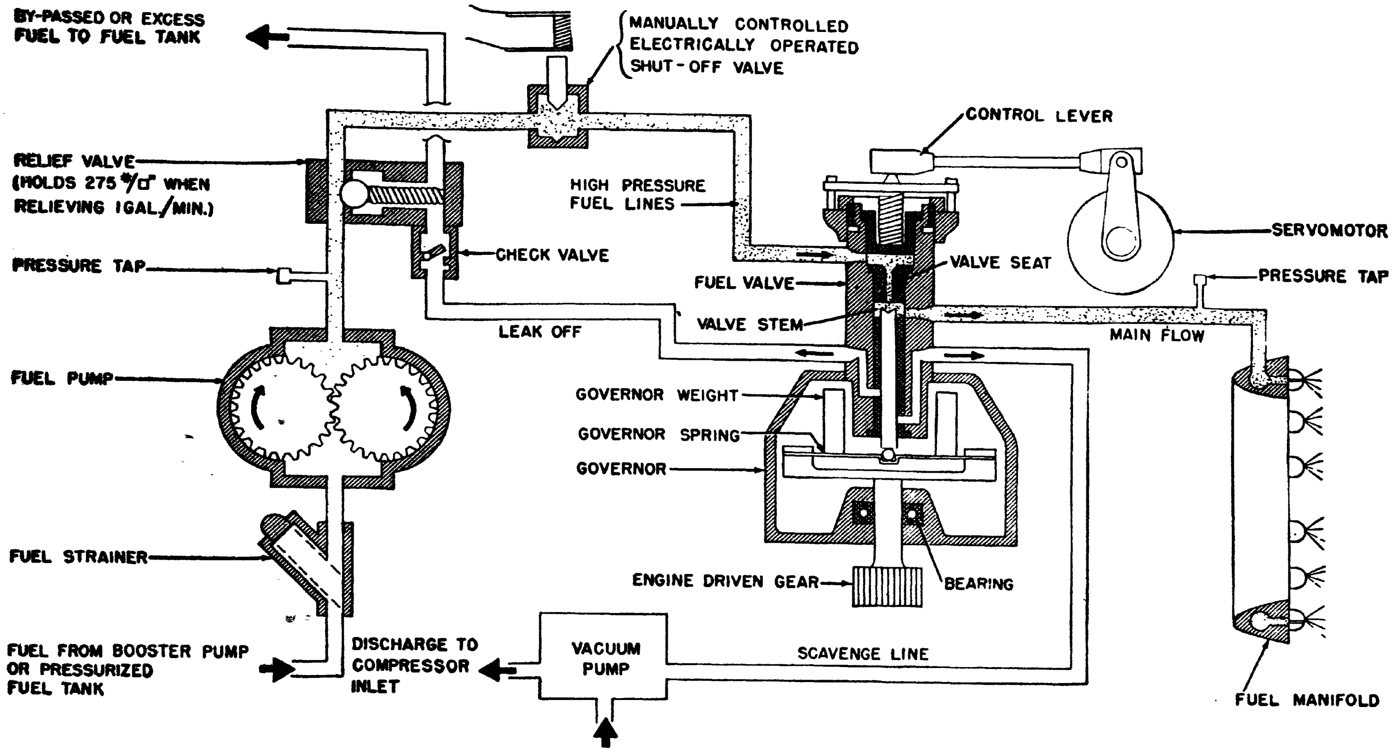

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Fig. 5. Schematic Drawing of Fuel Control System.

Air from the compressor enters the reservoir through a boss in the bottom of the cover and passes through a tube which is approximately *U*-shaped. The upper leg of the tube near the bend is drilled to vent air to the reservoir. The upper leg ends at a boss with a venturi-shaped opening through the cover to the exterior of the reservoir. From a hole drilled into the throat of the venturi, a tube leads to an oil filter strapped to the lower leg of the *U*-shaped tube. An oil level rod is screwed into a tapped hole in the top of the reservoir. A boss provided with a strainer is located near the top of the cover to provide a filling connection. Another boss at the top and at the bottom of the reservoir provides connections for a transparent tube to serve as an oil-level indicator.

Nozzles and Oil Sprays

The only special nozzle that is used sprays the lubricating oil-air mist into the turbine bearing in the combustion-chamber assembly. The mist which lubricates the spiral gear and pinion and the compressor spindle bearing in the bearing-support assembly issues from the open end of the tubing itself, and the mist which lubricates the gears in the gear-box assembly is discharged through an orifice nipple.

Lubrication-system Piping

The piping includes the vacuum-pump suction and discharge lines, the compressed-air line leading from the compressor to the oil reservoir, and the lubrication lines leading from the oil reservoir to the lubrication points (Figs. 2, 3, and 4).

Fuel-control System

Operation

The fuel-control system supplies atomized fuel to the combustion chamber and, by regulating the flow of the fuel, controls the speed of the engine and the thrust developed. Engine speed and thrust output increase when the fuel-control system is adjusted to give a higher rate of fuel flow, but the speed remains relatively constant for any given setting. A schematic diagram of the fuel-control system is shown in Figure 5.

To insure a satisfactory operation at high altitudes, fuel must be supplied to the fuel-control system by a booster pump or pressurizing system capable of delivering a minimum of 920 lb of fuel per hr at 5 to 25 psi (gauge). Depending on the installation, the fuel supply may be followed by a fuel shut-off valve to facilitate uncoupling the fuel-control system from the fuel supply during engine overhaul.

Fuel enters the fuel-control system through a strainer on the intake of the positive-displacement gear-type fuel pump. The fuel pump, being driven by the gear box, has a straight-line discharge characteristic and delivers fuel at a rate directly proportional to engine speed. The fuel-demand curve of the jet engine is parabolic, so the fuel pump has sufficient capacity to supply the engine over the entire operating range. A relief valve on the discharge side of the fuel pump by-passes the excess fuel back to the fuel supply, thereby maintaining a relatively constant pump discharge pressure. The relief valve is set to maintain a pressure in the fuel line of 275 psi when by-passing one gallon per min.

Fuel flows from the relief valve to the governor through a manually controlled, electrically operated, shut-off valve. The shut-off valve, used only in starting and stopping the engine, has two positions, *fuel on* and *fuel off*. The valve is located off the engine to suit the needs of any individual installation. Connections to and from the shut-off valve must be made with those that can withstand gasoline at high pressure (approximately 275 psi).

The governor regulates the rate of fuel flow from the shut-off valve to the nozzles in the combustion-chamber inlet casing. There, the fuel is atomized and sprayed into the combustion chamber. The governor serves as a throttle as well as governor. The setting of an arm on the top of the governor determines the approximate rate of fuel flow, and consequently, the engine speed. The spring-type centrifugal governor, driven by the gear-box assembly, maintains this fuel flow. If the engine speeds up or slows down, the fuel supply is decreased or increased by the governor until the engine resumes its former speed. Thus, the governor maintains a relatively constant en-

gine speed under all conditions of altitude and flight speed for any given governor-arm setting.

To prevent leakage of fuel down into the rotating members of the governor-and-fuel-valve assembly, a leak-off line is provided to carry such fuel through a check valve to the relief valve and back to the fuel supply. As a second precaution, another line leads from a point immediately below the leak-off line in the governor to the intake manifold of the vacuum pump that is the scavenger of the lubrication system.

The arm on the governor is positioned by an electric servo motor mounted on a bracket at the rear of the gear-box assembly.

Fuel Pump

The fuel pump is located on top of the right-hand forward area of the upper gear-box housing (Figs. 3 and 4).

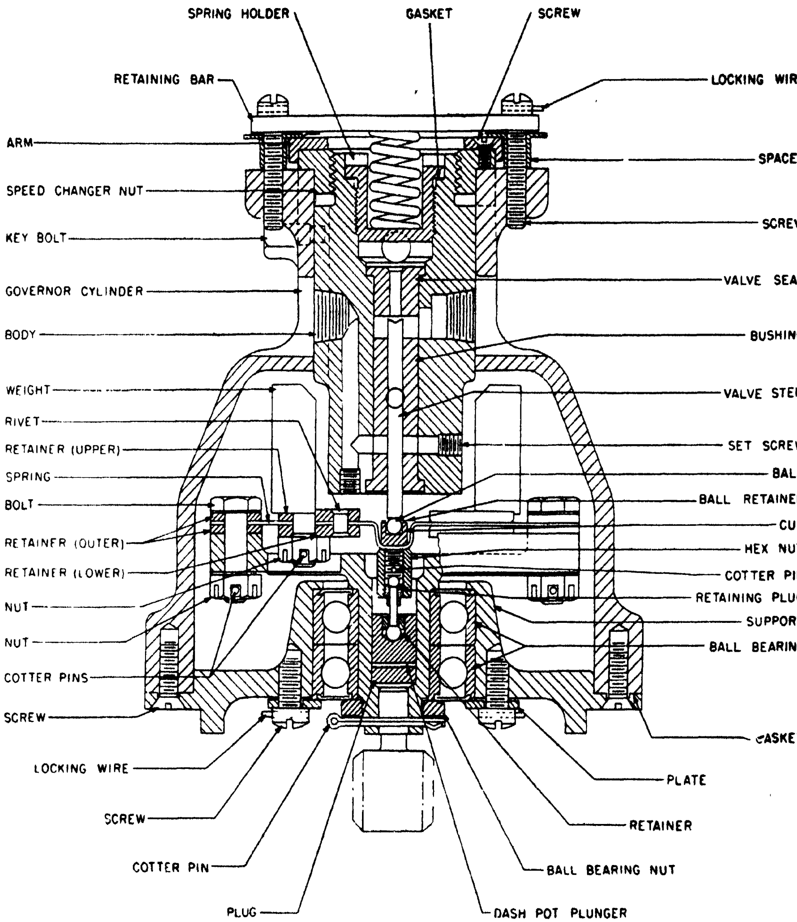
Governor

The governor-and-fuel-valve assembly is mounted on a pad (Figs. 3, 4, and 6). The assembly and pad are located on top of the rear right-hand area of the upper gear-box housing. The controlling elements of the assembly are a spring-type centrifugal governor and a two-part cup valve. Because the relief valve on the intake lines maintains a constant pressure fuel supply to the governor, the opening between the two parts of the cup valve determines the rate of fuel flow.

The upper half of the valve is positioned by an arm on the top of the governor. The arm is secured to, and rotates, a speed-changer nut. The valve body and seat are raised or lowered by the rotation of the speed-changer nut. A rod-like valve stem with a conical cup in the top extends vertically up through a bushing in the bottom of the body. The cupped end of the valve stem rests against the bottom of the valve seat and covers a vertical hole through the seat when the valve is closed. Fuel enters the body through a port above the valve seat, flows down through the hole in the seat, passes between the seat and valve stem, and leaves the body through another port. Raising or lowering the valve seat enlarges or diminishes

the opening between the valve seat and valve stem, and thus increases or decreases the rate of fuel flow through the valve seat.

The lower part of the cup valve, which consists of the ver-



Courtesy of Westinghouse Electric Corporation

Fig. 6. Cross Section of Governor.

tical valve stem extending up into the valve body, is positioned by the centrifugal governor. The spline drive of the governor, driven by the gear box, rotates a hub mounted in ball bearings. Across the top of the hub is a horizontal, flat spring supported at both ends. In the center of the spring are a cup and

ball which support the lower end of the valve stem. On top of the spring, about midway between the center and each end, a weight is secured. Because the governor is driven by the engine through the gear-box assembly, an increase in engine speed means a proportional increase in governor speed. The greater centrifugal force exerted on the governor weights causes the weights to lean farther away from the center of the flat spring. This movement of the weights bends the spring so that its center is raised. The valve stem, which rests on the center of the spring, is also raised and thus limits the fuel flow to hold the engine at its original speed. A dashpot arrangement attached to the center of the governor spring smooths out governor operation by opposing any sudden changes in valve setting.

When the throttle is opened, the governor arm moves clockwise and raises the valve body and seat, permitting an increased fuel flow between the valve seat and valve stem. When increased fuel flow obtained in this manner speeds up the engine, centrifugal action by the governor weights can only partially counteract the speed change, thereby enabling the engine to stabilize at a higher rate of speed.

Spiral grooves are cut in the cylindrical surface of the valve stem, and two holes are drilled through the bushing guiding the stem. Fuel that leaks down between the valve stem and bushing is carried out through the upper hole, which connects with the leak-off line. The second hole in the bushing, below the leak-off hole, is connected by piping to the lubrication-system vacuum-pump intake manifold. A small amount of air passing along the valve stem and into the vacuum-pump suction line scavenges any fuel that escapes the leak-off line. This arrangement prevents any fuel from reaching the rotating parts of the governor.

Servo Motor

The servo motor is mounted on a bracket secured to the top of the compressor immediately to the rear of the accessories (Figs. 3 and 4). The servo motor is coupled by a turnbuckle-type connecting lever to the governor arm. The

connections to the throttle control that operates the servo motor are described below in the section *Electrical System.*

Shut-off Valve

The electrically operated shut-off valve is controlled by a switch.

Fuel Relief Valve

The relief valve is mounted on the engine just below and to the right of the accessories (Fig. 4).

Check Valve

The check valve is located on the upper rear part of the relief valve to prevent fuel by-passed by the relief valve or fuel from the main fuel supply from passing up the leak-off line into the governor-and-fuel-valve assembly (Fig. 4).

Strainer

The strainer, located on the intake of the fuel pump, prevents the entrance of foreign matter into the fuel-control system from the fuel supply (Fig. 3).

Fuel Nozzles

There are 12 nozzles spaced evenly around the rear face of the fuel manifold ring in the combustion-chamber inlet casing. Manufactured by Monarch Co., they are No. F-80-PLP nozzles.

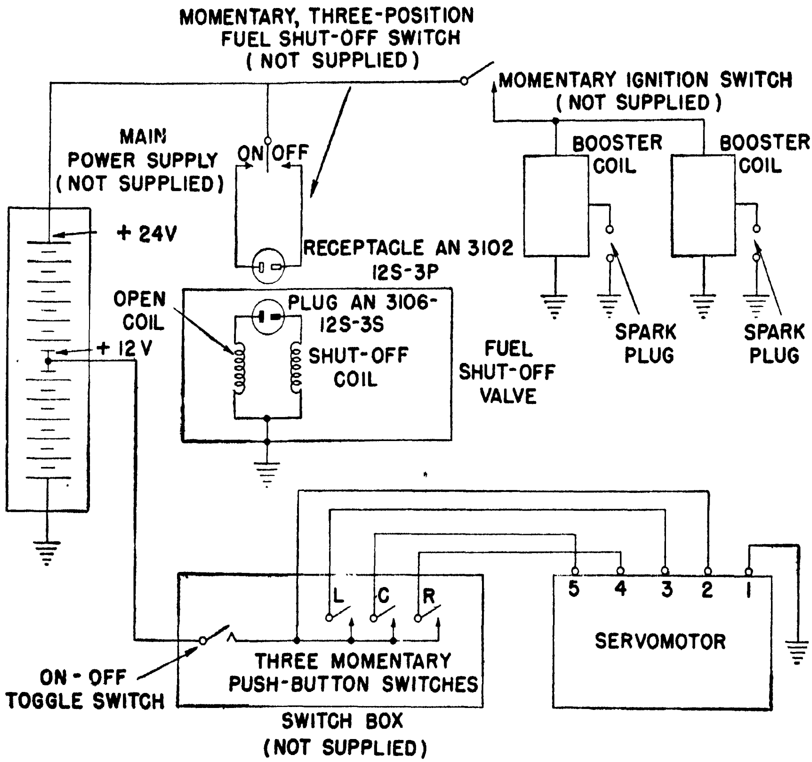
Fuel-control-system Piping

The piping includes the fuel-pump discharge line, the governor leak-off line, and the fuel supply line running from the governor-and-fuel-valve assembly to the combustion-chamber inlet casing (Figs. 3, 4, and 5).

Electrical System

The electrical system performs three functions: It provides current through the switch box to actuate the servo motor and thus control engine speed. Through the fuel shut-off switch, it

provides current to actuate the fuel shut-off valve. By means of the ignition-switch booster coils and spark plugs, it ignites the fuel-air mixture in the combustion chamber to initiate combustion. A schematic drawing of the electrical system is shown in Figure 7. Color codes are not used on any electrical-system wiring.



Courtesy of Westinghouse Electric Corporation

Fig. 7. Schematic Drawing of Electrical System.

The throttle-control section of the electrical system operates from a 12-v, d-c source. The switch box recommended for use as a throttle control contains a toggle switch, which energizes the servo-motor fields, and three momentary, push-button switches (*L*, *R*, and *C*, Fig. 7). Switch *L* causes the servo motor to move from any position toward a setting for minimum, idling engine speed. Switch *C* operates the servo motor toward a central position. Switch *R* causes the servo motor to

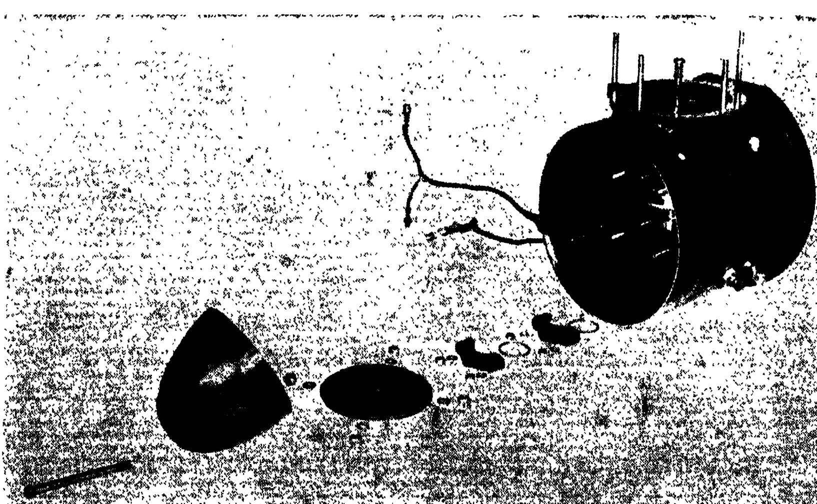
move from any position toward a setting for maximum engine speed.

The fuel shut-off section of the electrical system operates from a 24-v, d-c source. The switch recommended for use with the fuel shut-off valve is a momentary, three-position switch. This switch should have two separate contact positions and a neutral center position. Through the polarized receptacle, the switch activates either the coil that opens, or the coil that closes, the fuel shut-off valve.

The ignition section of the electrical system operates from a 24-v, d-c source. A momentary, push-button switch is recommended for use with the booster coils and spark plugs.

Front-bearing Support Assembly

The front-bearing support assembly, shown in Figures 8 and 9, is the forward section of the engine. The bearing support houses and supports the compressor bearing and reduction gearing which couple the compressor spindle to the gear box. The gear box, mounted on top of the bearing support,



Courtesy of Westinghouse Electric Corporation

Fig. 8. Front-bearing Support Assembly—Front View of Partially Disassembled Bearing Support.

contains additional reduction gearing to drive the accessories mounted on the gear-box housing.

The air required to support combustion enters the engine through an annular opening in the bearing support. Guide vanes at the rear of the assembly direct the air flow smoothly into the compressor assembly, which is the second section of the engine.

Bearing Support

The bearing support, or housing, is a magnesium casting which consists essentially of the concentric cylinders interconnected by three radial struts. Inside the rear of the inner cylinder is a seat for the compressor spindle bearing. A spiral gear and pinion, also in the inner cylinder, serve as reduction gearing to take power from the compressor spindle. The pinion is mounted in the end of the compressor spindle; the gear is mounted between ball bearings in the inner cylinder. A fairing cone and cover streamline and close the front end of the inner cylinder.

The three radial struts interconnecting the inner and outer cylinders are spaced 120 deg apart. The top strut provides access to the inner cylinder for the gear-box drive shaft. The other two struts house thermocouple leads and lubrication-system tubing.

The single-walled outer cylinder has a flange at the rear edge which is bolted to the compressor assembly, the following section of the engine. A mounting face for the gear box is cast on top of the outer cylinder.

The outer surface of the inner cylinder, the struts, and the inner surface of the outer cylinder are smoothly finished to prevent disturbance of the air taken into the engine.

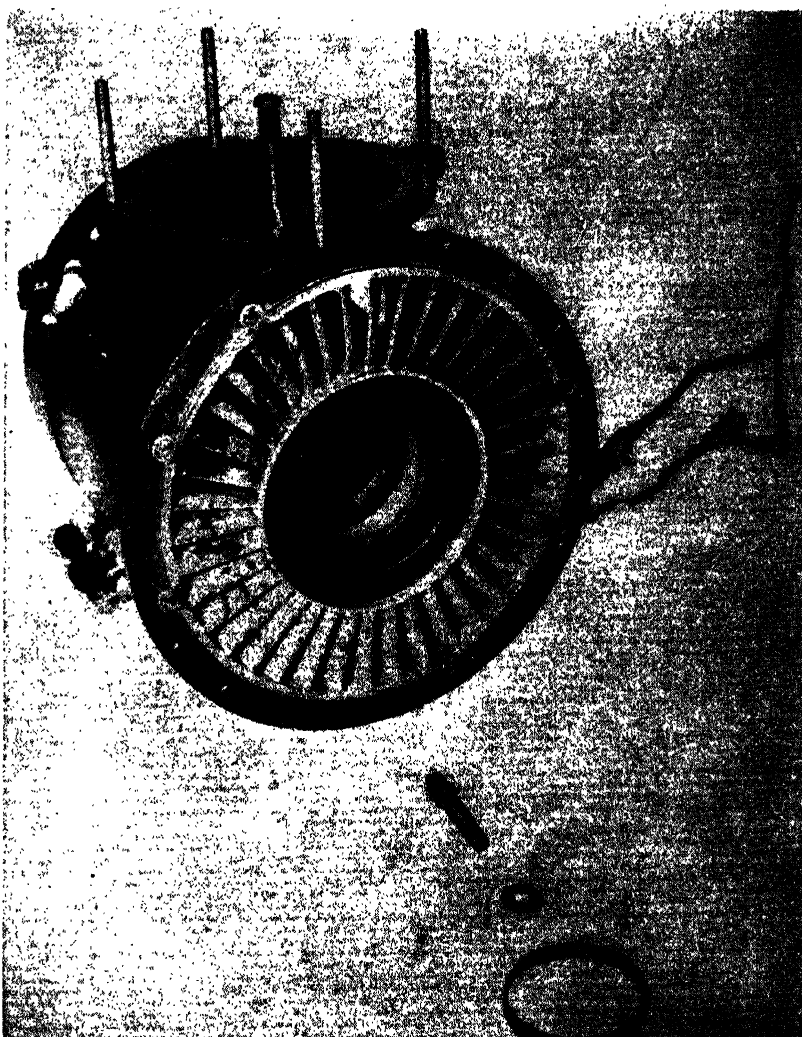
Inlet Guide Vanes

Between two cylindrical, concentric shrouds, 38 guide vanes are mounted radially (Fig. 9). Tongues on each end of the blades are welded into holes in the shrouds. The annular, inlet guide-vane subassembly fits between the inner and outer cylinder at the rear of the bearing support. The subassembly is

held in place by six screws in the rear flange face of the outer cylinder.

Accessory Gear Box

The accessory gear box is mounted on top of the bearing support and contains reduction gearing to drive four of the



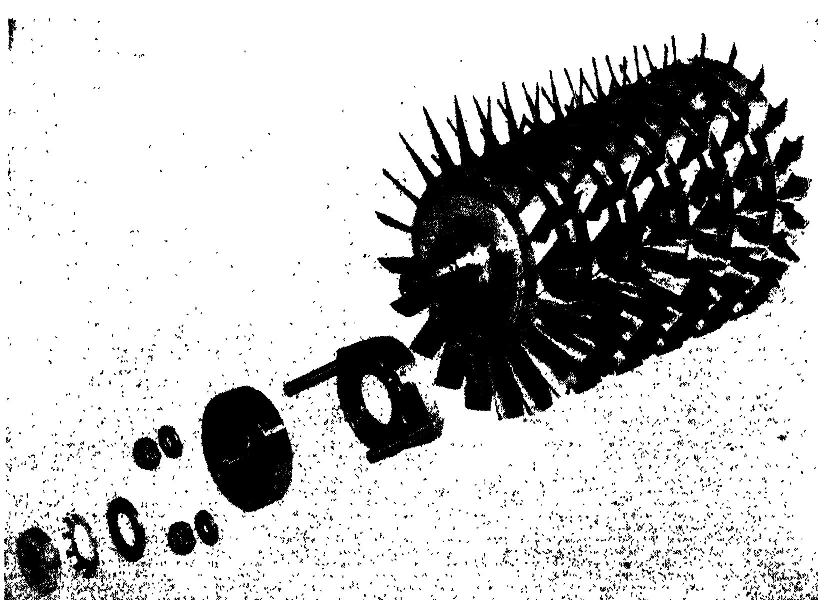
Courtesy of Westinghouse Electric Corporation

Fig. 9. Front-bearing Support Assembly—Rear View of Partially Disassembled Bearing Support Showing Inlet Guide Vanes in Place.

accessories: the vacuum pump, fuel pump, governor, and tachometer generator. A spline shaft extends through the bottom of the gear box and connects with the spiral gear in the bearing support. On top, the spline shaft fits into a pinion which drives four gear wheels mounted in ball bearings in the upper and lower gear-box housings. Each of these gears drives the shaft of one of the four accessories mounted on the gear box.

Compressor Assembly

The compressor assembly is bolted between the front-bearing support assembly and the suspension plate on the combustion-chamber assembly. Air, passing through the inlet guide vanes at the rear of the front-bearing support assembly, is compressed as it flows axially through the compressor. The outlet guide vanes at the rear of the compressor straighten the



Courtesy of Westinghouse Electric Corporation

Fig. 10. Compressor Assembly—Front View of Disassembled Compressor Spindle and Blade Subassembly.

air flow and direct it smoothly into the combustion-chamber inlet casing.

The compressor is a six-stage (six rows of rotating blades), axial-flow, constant-tip-diameter type. Rotating members of the compressor are all part of the spindle-and-blade subassembly, shown in Figure 10. Five rows of stationary blading which alternate with the rows of rotating blades are part of the compressor-diaphragm subassembly. Three rows of stationary blading at the rear of the compressor make up the compressor-outlet guide-vane subassembly. The upper and lower halves of the compressor diaphragm and compressor-outlet guide-vane subassemblies are secured to the compressor cover and base respectively, as shown in Figure 11.

Compressor-spindle-and-blade Subassembly

The compressor-spindle-and-blade subassembly may be divided into the compressor spindle, the rotating blades, and the bearing and attendant parts.

The compressor spindle is machined from a single aluminum-alloy forging. Diameters of the six rotor disk sections successively increase from the front to the rear of the spindle. The circumferences of the rotor disk sections are grooved to hold the ball roots of the rotating blades. In addition, a small hole is drilled radially in the bottom center of each blade groove to hold the tail of a locking key so that the key will not slide out of the groove. Shoulders project from the front and rear faces of the rotor disks at a radial distance slightly less than the bottom of the blade grooves. These shoulders and the seal strips of the compressor-diaphragm subassembly act as a seal to prevent the leakage of air between stages. A stub shaft on the front end of the spindle and a coupling flange at the rear end are integral parts of the spindle. The coupling flange has 12 equally spaced, drilled and reamed bolt holes to provide means for bolting the flange to a coupling on the turbine shaft. The face of the flange is machine-finished to close tolerances and has a male spigot with a recessed center to clear a projecting bolt in the center of the turbine shaft.

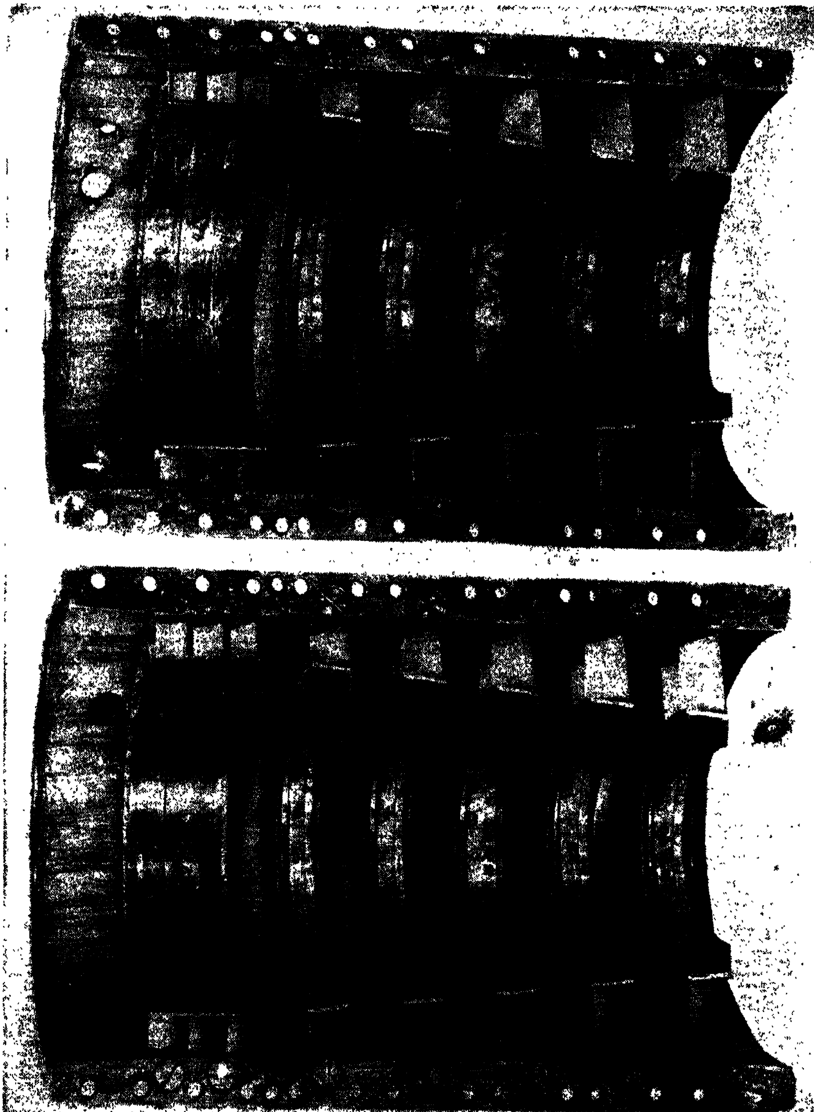
Each rotating blade is machine-finished to a double, con-

vex cross section. Rotating blades in the first stage are heavier than in other stages to withstand the higher stresses set up by air entering the compressor. Rotating blades in the second to the sixth stages are similar, but the over-all length of the blades is smaller in each successive stage so that the tip diameters of all the blades are equal. Each rotating blade is secured to the spindle by means of a ball root, which is grooved at the bottom to receive a locking key. After the locking key and blade are inserted in the spindle, the ends of the locking key are peened over the blade root and filled flush.

The compressor bearing, seated in the front-bearing support assembly, supports the forward end of the compressor spindle. The inner race of the bearing fits on a sleeve which is a shrink fit over the stub shaft on the forward end of the compressor spindle. The inner race of the bearing is secured to the spindle by means of a washer, lock washer, and lock nut. A ball bearing retainer which acts as a rub ring in case of bearing failure fits against the rear side of the bearing. By means of two studs, washers, and nuts, the outer race of the bearing is clamped into the bearing bore between a liner (which is part of the front-bearing support assembly) and the bearing retainer. The end of the stub shaft is drilled and tapped to receive a sleeve. The pinion of the front-bearing support assembly screws into this sleeve.

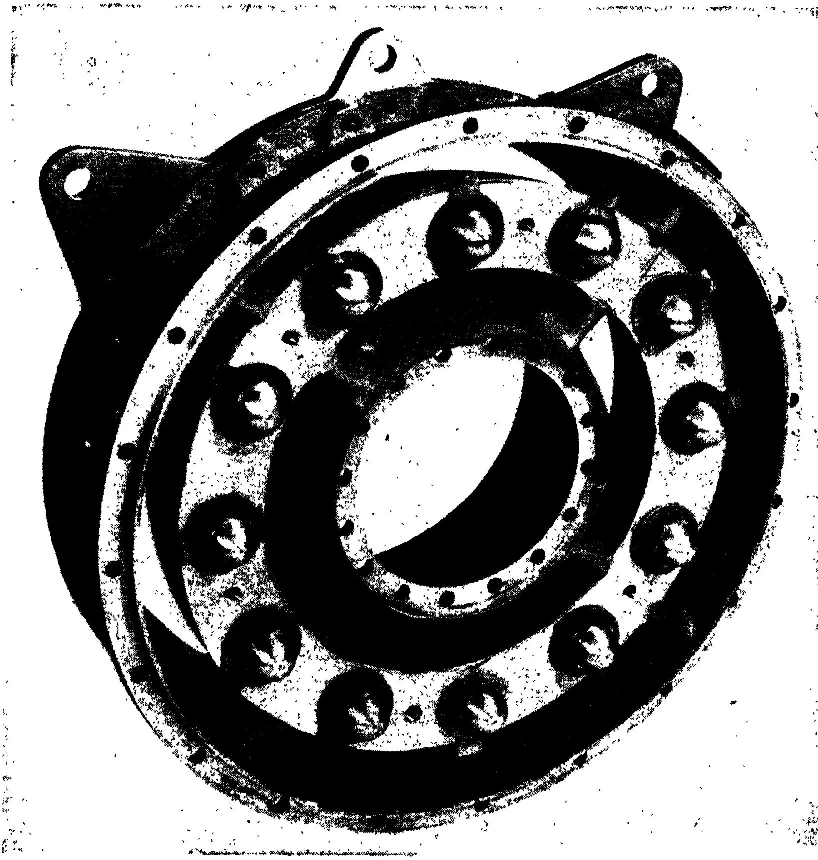
Compressor Cover and Base

The compressor cover and base are semicylindrical aluminum castings which, when bolted together at their horizontal flanges, form the compressor cylinder (Fig. 11). Internal surfaces of the cover and base are grooved to hold the five rows of stationary blading making up the compressor-diaphragm subassembly and the three rows of stationary blading making up the compressor-outlet guide-vane subassembly. Undercut faces on the front and rear vertical flanges of the compressor cover and base are accurately machined and scraped to mate with the joining faces of the front-bearing support assembly and the suspension plate on the front of the combustion-chamber inlet casing. The compressor cover and base are bolted



Courtesy of Westinghouse Electric Corporation

Fig. 11. Compressor Assembly—Interior of Compressor Cover and Base Showing Diaphragm and Outlet Guide-vane Subassemblies.



Courtesy of Westinghouse Electric Corporation

Fig. 12. Combustion-chamber Assembly—Rear View of Combustion-chamber Inlet Casing Showing Fuel Nozzles in Place.

Courtesy of Westinghouse Electric Corporation

Fig. 13. Combustion-chamber Assembly—Left Side View of Burner Shell, Inner Case, and Outer Case with Spark Plugs and Lubrication System Tube-to-turbine Bearing.

to these adjoining members and to each other. A single circumferential rib reinforces the cover and base. Drilled and threaded bosses are provided for mounting accessories and as a tap for compressed air used in the lubrication system.

Compressor-diaphragm Subassembly

The compressor-diaphragm subassembly is made up of five pairs of diaphragm halves (Fig. 11). In each half diaphragm is a row of machine-finished, stainless-steel, stationary blades held in position between semicircular, stainless-steel, inner and outer shrouds. Pegs on the top and bottom of the blades are welded into slots in the shrouds. Two semicircular seal strips with an *L*-shaped cross section are welded to the face of the inner shroud. These strips act as a seal to prevent the flow of air between the compressor diaphragms and the shoulders on the rotor disks of the compressor spindle. Each half diaphragm fits into a groove in the compressor cover or base and is locked in place by two screws in the horizontal flanges of the cover and base.

Compressor Guide-vane Subassembly

The compressor guide-vane subassembly is made up of three pairs of diaphragm halves. In each half diaphragm is a row of machine-finished, aluminum, stationary blades held in position between semicircular, aluminum, inner and outer shrouds. Pegs on the top and bottom of the blades are welded into slots in the shrouds. Half of each of the three diaphragms fit together into a single wide groove at the rear of the compressor cover or base. Each set of three diaphragm valves is locked in place by four screws in the horizontal flanges of the cover and base.

Combustion-chamber Assembly

Operation

The interconnecting cylinder and the combustion-chamber inlet casing act as a diffuser and transition section between the compressor and the main part of the combustion chamber. The inlet casing also serves as a fuel manifold. A tubular

stream of air from the compressor is divided into two concentric, tubular streams as it passes through annular openings in the combustion-chamber inlet casing. These two concentric streams flow into the space between the cylindrical outer and inner cases of the combustion chamber, shown in Figure 13. Separating the two concentric streams of compressed air are the perforated walls of the annular burner shell, also shown in Figure 13. The burner shell contains holes located in the inner and outer walls arranged in rows parallel to the axis of the engine. These holes, which are small at the upstream end and gradually increase in size toward the downstream end, serve to meter the air into the burner shell. The small upstream holes admit the primary air which mixes with the fuel in the proper amount for efficient combustion. The larger downstream holes admit the secondary air, the major portion of the total air flow, which serves to cool the combustion gases, thereby bringing them to a usable temperature. The resultant mixture passes through stationary guide vanes of the turbine nozzle and drives the rotating turbine blades. Exhaust gases continue to the rear through the exhaust-nozzle assembly, which is the last main section of the engine.

To start the engine under static conditions, compressed air from an external source is supplied to an air manifold at the rear of the combustion chamber. The air is directed against the turbine blades to rotate the turbine shaft until starting speed is attained.

The most forward component of the combustion chamber is the interconnecting cylinder, which projects into the rear of the compressor assembly. This is followed by a suspension plate mounted on the front flange of the combustion-chamber inlet casing. On the rear flange of the inlet casing, each inside the other, are mounted the outer case, burner shell, and inner case. The annular turbine nozzle is fitted between the inner and outer cases at the rear of the burner shell. The turbine shaft is supported by a bearing in the inner case. The shaft extends through the center of the entire length of the combustion-chamber assembly, from the coupling on the compressor

spindle to the rotating turbine blades housed in the exhaust-nozzle assembly.

Intercooling Cylinder

The interconnecting cylinder forms the inner surface of the annular opening leading from the rear of the compressor to the combustion-chamber inlet casing. Divided horizontally into an upper and lower half, the cylinder is flanged around its inner surface at the rear to fit around a mating flange and groove on the combustion-chamber inlet casing. Each half is secured to the inlet casing by three screws and the two halves are bolted together. Circular projections on the forward edges of the interconnecting cylinder act as a seal to prevent the leakage of air between the cylinder and the rear rotor disk face of the compressor spindle.

Inlet Casing

The combustion-chamber inlet casing is made by casting aluminum alloy around a prefabricated stainless-steel fuel manifold. The major part of the subassembly consists of three concentric cylinders interconnected by four radial struts as shown in Figure 12.

The flanged outer cylinder has a single wall that diverges toward the rear. The flange on the front end has an undercut face to mate with a lip on the suspension plate and is bolted through the suspension plate to the compressor assembly. The flange on the rear end of the outer cylinder has a lip that fits into and bolts to the outer case of the combustion chamber.

The inner cylinder has a single wall that converges toward the rear. A channel around the outside of the front end provides support for the interconnecting cylinder. A boss at the rear of the inner cylinder slides around and secures the front flange on the inner case of the combustion chamber.

The middle cylinder, suspended between the inner and outer cylinder by four struts, is roughly triangular in cross section and contains a fuel passage. Fuel is supplied to the interior of the middle cylinder through one of the struts. Thermocouple wiring and lubrication-system tubing pass through

the other struts. Spaced evenly around the rear face of the middle cylinder are 12 holes drilled through to the fuel passage in the cylinder, each hole being tapped for a fuel nozzle. The middle cylinder also serves as a support for the burner shell.

Outer Case

The stainless-steel outer case is fabricated from forgings and sheet stock by resistance welding (Fig. 13). A flange, welded on the front end of a single-walled outer cylinder, mates with, and is bolted to, the outer ring of the combustion-chamber inlet casing. Inside the rear end of the outer cylinder is welded the outer nozzle ring. This ring has a cross section which is roughly *U*-shaped and forms a sealed passageway which runs circumferentially around the inner surface of the outer cylinder. A conical section, the deflection cone, provides a smoothly constricting inner surface between the outer cylinder and the forward edge of the outer nozzle ring. Near the forward edge of the outer cylinder are two bosses to provide access through the cylinder for two spark plugs. Around the inside of the cylinder, approximately three quarters of the length to the rear, 16 equally spaced guide buttons are spot-welded. These buttons support the outside of the burner-shell assembly. Just to the rear of these buttons is a boss that gives access through the outer cylinder and the deflection cone for a pressure rake. Through another boss in the outer cylinder adjacent to the outer nozzle ring, compressed air for starting the engine is supplied to the circular passage formed by the flange ring. Eight holes through the inner wall of the ring mate with similar holes in the turbine-nozzle assembly. Through these holes, the compressed starting air is directed against the rotating turbine blades until starting speed is attained. On the rear of the outer nozzle ring are bolted the exhaust nozzle and back-pressure control assembly.

Inner Case

The stainless-steel inner case is fabricated from forgings and sheet stock by resistance welding (Fig. 13). A front flange

welded onto the single-walled bearing-support cylinder mates with, and is bolted to, the inner cylinder of the combustion-chamber inlet casing. The inner nozzle ring, on which the turbine nozzle is mounted, is welded at the rear around the outside of the bearing-support cylinder. This ring has a cross section which is roughly T-shaped. A conical section, the deflection cone, provides a smoothly diverging outer surface between the bearing-support cylinder and the forward edge of the inner nozzle ring. The turbine shaft runs through the bearing-support cylinder. Inside this cylinder, approximately at the center, is the cylindrically shaped bearing housing. The bearing on the turbine shaft rides in a bearing sleeve which, in turn, is held in the bearing housing by a bearing retaining ring. Lubrication-system tubing runs from the combustion-chamber inlet casing through the bearing-support cylinder to the turbine bearing. Around the outside of the bearing-support cylinder, slightly to the rear of the bearing support, eight equally spaced guide buttons are spot-welded. These buttons support the inside of the burner shell.

Burner Shell

The major items in the burner shell are two perforated, concentric, roughly cylindrical, sheet-metal shells. The inner shell converges for about three quarters of its length and then diverges. The outer shell diverges for the same distance and then converges. As a result, the cross-sectional area between these shells increases for most of their length and then decreases. Reinforcing wire rings are spot-welded to both the inner and outer shells. A flanged ring, welded between the inner and outer shells at the front, is bolted to the middle cylinder of the combustion-chamber inlet casing. A ring-shaped inner and outer cone reinforce the rear edges of the inner and outer shell respectively.

Turbine Nozzle

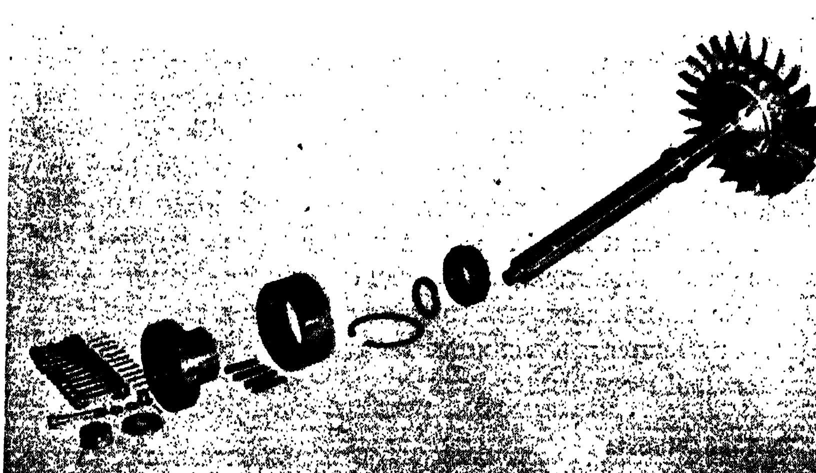
Between two cylindrical, concentric shrouds, 62 nozzle vanes are mounted radially. The vanes are a free, sliding fit in slots in the shrouds to permit differential expansion. The

annular turbine nozzle fits between the inner and outer nozzle rings at the rear of the inner- and outer-case assemblies respectively. Four retaining plates screwed to the outer nozzle ring secure the outer shroud, and four lock-wired tap bolts secure the inner shroud to the inner nozzle ring.

Turbine Shaft-and-blade Subassembly

The turbine shaft-and-blade subassembly may be divided into the turbine shaft, the rotating blades, the coupling and attendant parts, and the bearing and attendant parts (Fig. 14).

The turbine shaft and rotor disk are machined from a single forging. The circumference of the rotor disk is grooved to hold the ball roots of the rotating blades. The front end of the shaft is cut away to leave a threaded stub protruding. To the rear of the stub, the shaft is chamfered and slotted to receive four locking keys. At about two thirds of the distance from the front end of the shaft, a shoulder on the shaft provides a rear support for the inner race of the turbine bearing. A shrink ring fits in a groove in the shaft to support the front face of the inner race.



Courtesy of Westinghouse Electric Corporation

Fig. 14. Combustion-chamber Assembly—Front View of Disassembled Turbine Shaft-and-blade Subassembly.

The rotating blades, by a special process, are cast to size. The blade roots are shaped by machine. The concave-convex blades are driven into place, but are not a taper fit in the rotor disk. A notch is cut in the side of the shoulder on the blade root. Metal on the circumferential face of the rotor disk is peened into the notch to hold the blade firmly.

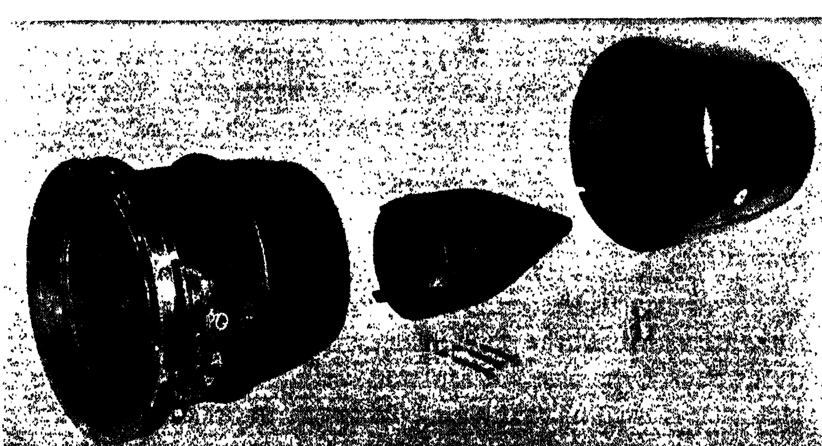
A flanged coupling slides over the front end of the shaft. Four keys fit into keyways in the coupling and shaft to prevent rotation between the shaft and coupling. A washer, castle nut, and cotter pin secure the coupling on the tapered portion of the shaft. The coupling flange on the front end of the shaft is bolted to the coupling on the compressor spindle.

The turbine bearing is a Hyatt roller bearing. It supports the turbine shaft and is mounted in the bearing housing of the inner case. The bearing is lubricated as described in the section *Lubrication System.*

Exhaust-nozzle Assembly

Operation

Exhaust gases from the turbine of the combustion-chamber assembly flow through an annular opening in the exhaust-



Courtesy of Westinghouse Electric Corporation

Fig. 15. Exhaust-nozzle Assembly—Left Side View of Discharge Case, Inner Case, and Fairing Cone.

nozzle assembly (Fig. 15). The inner surface of this opening is formed by the barrel and the inner cone; the outer surface, by the discharge case. In the opening, the velocity of exhaust gases is increased by expansion before the gases are jetted out the rear of the engine.

The exhaust-nozzle assembly may be divided into four main parts: The cylindrical barrel is supported inside the cylindrical discharge case by three, equally spaced radial struts. The inner cone slides into and projects from the rear of one barrel. The fairing cone slides over and streamlines the exterior of the discharge case.

Discharge Case

The discharge case is nearly cylindrical, although the body converges slightly toward the rear. A flange ring is welded around the front end of the body and a stiffener angle consisting of a hoop with an *L*-shaped cross section is welded around the middle of the body. The flange ring is drilled so that it can be bolted to the outer nozzle ring at the rear of the compressor assembly. Starting at the top center of the body, three equally spaced struts are welded radially to the interior of the body. These struts enclose three bolts which extend from the exterior of the discharge case into the inner cone to secure the cone in position. A boss on the left side below the center of the discharge case provides access through the case for a pressure rake.

Barrel

The barrel is a short cylinder welded to the inner ends of the three struts in the discharge case. A dished head closes the front end of the barrel except for a small hole in the center. The barrel is supported close behind the turbine rotor of the combustion-chamber assembly. The hole in the dished head allows the stub shaft on the turbine spindle to project into the barrel.

Inner Cone

The forward third of the inner cone is a cylinder. Welded to, and closing, the rear of the cylinder is a roughly conical, streamlined tail piece. Inside the front and rear of the cylindrical section are welded stiffener rings with *L*-shaped cross sections. Three lock bars, strips of reinforcing metal, are spaced 120 deg apart and welded longitudinally on the inside of the cylindrical section. A series of holes are drilled and tapped through each lock bar and the inner cone. The inner cone may thus be inserted varying distances into the barrel and secured in position by bolts passing through the radial struts in the discharge case and screwing into any of the holes in the lock bars.

Fairing Cone

The fairing cone is roughly cylindrical and converges slightly toward the rear. The cone slides over and streamlines the exterior of the discharge case. A reinforcing ring is welded inside the front edge of the fairing-cone body. Through the ring and body, eight holes are drilled and countersunk so that the fairing cone may be screwed to the flange ring around the front of the discharge case. A hole on the left side, below the center of the body, provides access through the fairing cone for a connection to the pressure rake in the discharge case.

Accessories

An accessory gear box is provided at the top of the engine for the four engine-driven accessories. A servo-motor control for the governor speed changer is also included as a standard accessory.

The following is a list and a weight breakdown of the complete engine and its required accessories.

Weight of basic engine	105.6 lb
Weight of basic engine and accessories	143.5 lb

Name	Weight in Lb
Gear box	9.5
Oil pump	1.4
Fuel pump	3.2
Fuel shut-off valve	2.4
All-speed governor	4.1
Fuel relief valve	0.7
Ignition coils (2)	3.3
Fuel strainer	0.3
Check valve	0.1
Spark plugs (2)	0.3
Oil-air mixer	0.3
Servo motor	5.0
Piping, fitting, and wiring	4.4
Tachometer generator	2.9
Total weight	37.9

Tachometer Generator

The tachometer generator is mounted on a pad. The generator and pad are located on top of the rear left-hand area of the upper gear-box housing. Capable of driving two indicators, the tachometer generator is designed for a top speed of 1700 rpm and a ratio of tachometer generator to engine rotor speed of 1 to 20. When tachometer indicator is used, the indicator will read $\frac{1}{10}$ engine speed.

Vacuum Pump

The vacuum pump is mounted on a pad, and is located on top of the forward left-hand area of the gear-box housing. The pump weighs 3.4 lb, has a displacement of seven cubic inches, and is designed for a top speed of 4000 rpm with a ratio of pump to engine rotor speed of 2 to 17. Working against an inlet low pressure of 1.96 psi (four inches mercury) below gauge and an outlet pressure of 0.48 psi (one inch mercury) above gauge, pump capacity at 1500 rpm is 4.5 cfm. The vacuum pump is provided with a shear section which breaks at a torque of 600 in.-lb to protect the engine from damage in case the pump is jammed.

Fuel Pump

The fuel pump is of the positive-displacement, gear type, is mounted on a pad, and is located on top of the right-hand forward area of the upper gear-box housing. With a ratio of

pump to engine rotor speed of 6 to 85, the pump delivers 635 lb of fuel per hr while running at a top speed of 2400 rpm. Capable of rotating in either direction, the pump weighs 2.7 lb, has a displacement of 0.227 cu in., and delivers a maximum, continuous, operating pressure of 1500 psi.

Servo Motor

The servo motor is a 12-v, d-c, compound-wound, electric motor. Mounted on brackets immediately to the rear of the gear-box housing, the servo motor is connected to, and regulates, the governor by a turnbuckle-type connecting lever. Drawing a maximum of 1.60 amp at full load, the servo motor delivers 75 in.-lb torque at a speed of 1.5 rpm. The servo motor weighs 3 lb 10 oz and is provided with class *A* insulation on all wiring.

Shut-off Valve

The shut-off valve is located off the engine to suit each individual installation. The valve is semibalanced and has an acceleration factor of 15 g. Maximum operating pressure is 400 psi. The valve passes fuel at the rate of 1.5 rpm with a maximum pressure drop of two pounds. Designed for intermittent duty and provided with separate open and shut-off coils, the valve draws 250 w from a 24-v, d-c source while opening or closing. An electrical receptacle and plug are supplied to connect the valve with its control switch.

Relief Valve

The relief valve is designed to handle gasoline over a pressure range of 200 to 300 psi. Mounted on the right side of the engine just below the gear-box housing, the valve must be tested in the field, and adjusted if necessary, to hold a pressure of 275 psi while relieving at a rate of one gallon per minute.

Check Valve

The check valve is located on the upper rear port of the relief valve. The bursting pressure of this valve is 1500 psi.

Fuel Strainer

The fuel strainer is located on the intake port of the fuel pump, and is provided with a 120-mesh screen.

Booster Coils

The two booster coils are located off the engine to suit each individual installation, and are designed for intermittent operation and draw 1 to 1.2 amp from a 24-v, d-c source.

Spark Plugs

The two spark plugs are located on each side of the lower front section of the combustion chamber. They are used only to initiate combustion.

Prerun Checks

Immediately before the jet engine is put into operation, a careful inspection should be made to insure against damaging the engine and surrounding equipment or injuring personnel.

Pressure-rake Alignment

If pressure rakes are used, the scribe-mark on the hex nut around each rake must be toward the front of the engine so a full pressure reading will be given.

Inner-cone Setting

The inner cone of the exhaust-nozzle assembly should not slide forward or to the rear under firm hand pressure.

In addition to scribe marks at ½ in. intervals on the inner cone, there is a scribe mark indicating proper inner-cone setting for engine operation at military rating. Unless the engine log book specifies that a different inner-cone setting should be used, the scribe mark indicating military position of the inner cone must line up with the extreme rear edge of the exhaust-nozzle fairing cone.

If the inner cone is loose or is not positioned correctly, set the cone.

Compressor and Turbine Clearance

Rotate the turbine slowly by sliding a soft rod, such as a wooden pencil, into the rear of the exhaust-nozzle assembly and pushing the rotating blades of the turbine in a counterclockwise direction. Listen carefully for blade rubs throughout the compressor assembly and around the turbine.

Ignition-spark Intensity

Remove the lead from one of the spark plugs in the forward lower section of the combustion-chamber assembly and hold the terminal of the lead approximately ½ in. from the side of the combustion chamber. Take all necessary safety precautions to avoid getting a shock or starting a fire. Push the ignition switch and observe whether a spark jumps the gap between the terminal of the lead and the side of the combustion-chamber assembly. Repeat the procedure with the lead to the other spark plug. A ½ in. ignition spark should be obtained.

Lubrication System

Unscrew the oil-level rod in the top of the oil reservoir, pull out the rod, and note the oil level as indicated by the length of rod covered with oil. When full, the oil reservoir provides lubrication for up to one hour of normal engine operation. If the reservoir is less than half full, or if engine runs of longer than 20 min are to be made, fill the reservoir as follows. Remove the plug from the elbow on the upper left-hand side of the oil-reservoir cover plate. Slowly fill the elbow with the proper *clean* oil. When the oil level can be seen in the elbow, the reservoir is full. *Do not fill the oil reservoir through the hole for the oil-level rod.* Securely replace the plug and the oil-level rod.

Unscrew the nut and disconnect the tubing from the elbow on the bottom of the oil-reservoir cover plate. Connect the elbow to an external source of clean, dry, compressed air. When air is supplied to the oil reservoir at a pressure of 30 psi, an oil-air mist should be evident inside the exhaust-nozzle assembly. There should be no evidence of leakage at any of the lubrication-system connections or fittings.

Operation of Throttle Control

Close the toggle switch that energizes the servo motors (Fig. 10). Hold momentary switch (*L*) closed until the servo motor stops. Whether or not the servo motor operates, note the position of the governor arm. Then hold momentary switch (*C*) closed until the servo motor stops and note the new position of the governor arm. Hold momentary switch (*R*) closed until the servo motor stops and note the third position of the governor arm. Again hold momentary switch (*L*) closed until the servo motor stops. Switch (*L*) should cause the servo motor to move the governor arm to an extreme counterclockwise position; switch (*C*), to a central position; switch (*R*), to an extreme clockwise position. Upon completion of inspection, open the toggle switch that energizes the servo motor.

Fittings and Wiring

Make a general inspection of all tubing, fittings, and connections for leaks; and of all wires, connectors, and terminals for defects.

Vacuum-pump Intake Port

If the vacuum-pump intake port is not connected to the engine accessories or to the oil-scavenge line from the front-bearing support assembly and accessory gear box, the port should be closed by a bleeder. This may consist either of a relief valve set for 3.4 psi (seven inches mercury) or an orifice with a $\frac{5}{64}$ in. opening.

Fuel-supply Pressure

Make sure the fuel shut-off valve is closed. Pressurize the fuel supply. Check the fuel pressure at the fuel-pump intake. If this pressure is not between 6 and 20 psi, check the fuel-supply boost pump or pressurizing system and correct the trouble.

Starting Air Supply

If the engine is to be started on the ground, a clean, dry supply of compressed air for starting the engine must be provided. The nose and fittings from this supply must have a

minimum ID of one inch. Approximately five pounds of air at 100 psi are required for each start. Check the starting air supply for connection through an air throttle valve to the elbow at the upper, right-hand, rear of the combustion-chamber assembly. Test the connections for leakage with a short burst of starting air.

Operating Procedure

The jet engine is designed to be started either on the ground by means of an auxiliary compressed-air supply or in flight by utilizing the ram action of air entering the air intake. Although the operating procedure is simple, no one should attempt to operate the engine until he is thoroughly familiar with all the engine controls and instruments and understands the operating precautions which must be observed to avoid danger to personnel and damage to the engine. *Dead-band* operation, caused by too rapid acceleration in the lower speed ranges, fuel flooding during starting, failure to extinguish internal flames immediately after the fuel shut-off valve has been closed, and other similar operational problems, if not corrected immediately, will seriously damage the engine.

When the engine has been taken from storage, a trial run must be made before the unit is put into general operation. For the trial, the engine should be run for 15 min at a speed of about 19,000 rpm. Do not exceed 21,000 rpm on a trial run. At the end of the trial, clean all strainers and filters and refill the lubricating system with new oil.

Starting Instructions

Starting in Flight by Ram

Always make a ground run before flight to check for leaks and loose connections.

The procedure for starting the engine in flight by ram is identical with that used on the ground except that the engine should be brought to a speed of 4000 rpm before ignition. Increased ram action takes the place of increased starting air pressure.

Starting by Compressed Air

This operation is accomplished step by step as follows:

Operation	Check	Precaution
Pressurize the fuel supply	Fuel-pump inlet pressure should be 6 to 20 psi	
Close toggle switch that energizes servo motor	Servo control should be in minimum idling position	
Bring engine to 4000 rpm with air throttle valve	Fuel-pump discharge pressure should be 258 to 300 psi	If pressure is incorrect, close air throttle valve and check relief-valve setting
Open fuel shut-off valve	If fuel-pump discharge pressure drops to between 75 to 100 psi, proceed with next stop immediately	If pressure does not drop, close air throttle *immediately* to prevent fuel flooding, and check fuel shut-off valve
Push ignition switch until ignition takes place	Ignition may be seen if mirror is placed to show interior of exhaust nozzle A dull popping sound indicates ignition Ignition will be shown by tachometer as speed increase Ignition will cause rapid rise in turbine outlet temperature	If ignition does not take place in 15 sec, release ignition switch and close air throttle Blow out excess fuel by driving engine on compressed air at 2000 rpm for at least 30 sec If engine fails to start in six successive tries, allow 10 to 15 min for ignition coils to cool
Upon ignition, open air throttle until engine reaches 15,000 to 18,000 rpm, then close air throttle	If engine does not reach 15,000 rpm with servo at minimum setting, tap servo full- or half-speed switch until engine reaches 15,000 rpm	Turbine outlet temperature must never exceed 1500° F
Tap servo half-speed switch once or twice a second until engine reaches cruising speed (29,000 rpm)	Fuel burning in exhaust nozzle indicates dead-band operation Dead-band operation may sometimes be heard as a roaring sound	Engine *must not* operate in dead-band condition Immediate and rapid acceleration usually brings engine out of dead band
Inspect tubing on engine for leaks		If fuel- or oil-line leaks are noticed, the engine should be shut down immediately An air leak may be temporarily ignored if sufficient pressure is maintained on oil reservoir

Operating Instructions

Once the engine has reached cruising speed, or about 29,000 rpm, there is no difference between operating the engine on the ground or in flight.

Under no condition should the engine be operated at military rating for runs of longer than one hour.

The three switches on the servo switch box operate the servo motor which, in turn, changes the position of the governor arm. Engine speed will be steadily increased while the full-speed switch (*R*, Fig. 10) is held closed until maximum engine speed is reached. Engine speed will either increase or decrease while the half-speed switch (*C*, Fig. 10) is held closed until the engine reaches cruising speed. Engine speed will steadily decrease while the low-speed switch (*L*, Fig. 10) is held closed. Care should be taken when using the switch not to decelerate below 15,000 rpm.

There is danger of dead-band operation from too rapid acceleration between the speeds of 15,000 and 25,000 rpm. Once cruising speed is attained, acceleration may be as rapid as desired, provided that turbine outlet temperature never exceeds 1500° F.

When changing engine speeds, keep a close check on temperature and pressure readings. Watch bearing temperatures until they are constant for a two-minute period. During relatively constant-speed operation, instruments should be checked every three or four minutes. The following temperatures and pressures are indicated during normal operation. If incorrect

Cruising-speed Readings

Turbine outlet temperature	1000 to 1200° F
Fuel-pump discharge pressure	250 to 300 psi
Air pressure to oil reservoir	18 to 20 psi
Compressor spindle-bearing (No. 1) temperature rise	30 to 70° F
Turbine-bearing (No. 2) temperature rise	190 to 236° F

Military-rating Readings

Turbine outlet temperature	1170 to 1250° F
Fuel-pump discharge pressure	250 to 300 psi
Air pressure to oil reservoir	20 to 30 psi
Compressor spindle-bearing (No. 1) temperature rise	75 to 100° F
Turbine-bearing (No. 2) temperature rise	275 to 300° F

readings are noted, the engine should be shut down immediately and the trouble rectified.

If the engine is run in a test cell set up so that observations inside the exhaust nozzle can be made, the temperatures of turbine hot spots should be checked with a pyrometer. At no time should the temperature of a hot spot exceed 1800° F.

Stopping Instructions

Emergency Shutdown

For emergency stops, close the fuel shut-off valve and extinguish the combustion in the engine with CO_2 before the engine stops running. If the shutdown takes place while in flight, ram air will extinguish the combustion. Do not start the engine again for at least five minutes after shutdown.

Normal Shutdown

To shut down the engine normally, hold the low-speed switch on the servo switch box closed until the engine decelerates to idling speed. Close the fuel shut-off valve and immediately open the throttle to drain excess fuel by holding the full-speed switch closed. In flight, ram air will extinguish the combustion. On the ground, combustion should be extinguished with CO_2 while the engine is still coasting at about 2000 rpm. The combustion must be extinguished *immediately* upon shutdown to avoid severe temperature stresses which may damage the engine. When the engine has coasted to a stop, hold the low-speed switch closed until the servo control reaches minimum position. Then open the toggle switch that energizes the servo motor. Do not start the engine again for at least five minutes after shutting down.

Postrun Checks

As soon as operation is discontinued, check the following to be sure the engine has been properly shut down and is ready to resume operation when required.

Fuel Shut-off Valve

The *fuel shut-off valve* must be closed or be put in the *off* position.

Servo Control

The *servo motor* should be in minimum idling position (governor arm in extreme counterclockwise position). If necessary, hold closed the low-speed switch (L) on the servo switch box until the governor arm is set correctly. The toggle switch that energizes the servo motor must be left open.

Starting Air Supply

If a *starting air supply* is used, the air throttle must be fully closed. If a valve is provided, the air pressure should also be cut off from the air throttle.

Fuel-supply Pressure

If a *booster fuel pump* or other means is used to pressurize the main fuel supply, the pump or pressurizing system must be turned off. Release the pressure on the main fuel supply.

Chapter XV

General Electric P-80 Power Plant, J-33

Lockheed P-80 Shooting Star

One of the world's fastest airplanes, the jet-propelled P-80 Shooting Star, is powered by the General Electric Model J-33 Gas Turbine. Its ceiling is well above 45,000 ft, a mile higher than the rated top altitudes of first-line reciprocating-engine fighters.

The Shooting Star [1] has a wing span of 38 ft, 10½ in.; an over-all length of 34 ft, 6 in.; and a height of 11 ft, 4 in. Its total weight empty is approximately 8000 lb. Gross take-off weight with maximum fuel capacity is about 14,000 lb, which is 4000 lb lighter than the gross weight of the conventional P-38 Lightning.

The U.S. Army Model P-80A-1 Airplane, the official designation of the Shooting Star, was designed and manufactured by Lockheed Aircraft Corporation, Burbank, Calif. It is powered by the gas turbine unit described in this chapter.

Models J-33-9 and J-33-11 turbo-jet units (General Electric model designations I-40-9 and I-40-11) are basically similar. Certain differences between models, such as the arrangement of external piping and accessories, will not be discussed. Throughout this chapter, the unit will be referred to as the Model J-33.

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[1] Both the plane and power unit were developed under tremendous wartime pressure at great speed by teamwork of the Air Technical Service Command, U.S. Army Air Forces, and engineers of Lockheed Aircraft Corporation and General Electric Company. Clarence L. Johnson, Chief Research Engineer for Lockheed, designed and supervised construction of the first Shooting Star in 143 days. Reginald G. Standerwick, General Electric engineer, had charge of design and development of this powerful aircraft gas turbine.

This chapter contains a description of, and operating and maintenance instructions for, the Models J-33-9 and J-33-11 turbo-jet engines designed and manufactured by the General Electric Company, Schenectady, N. Y., and also manufactured by the Allison Division of General Motors Corporation, Indianapolis, Ind.

The definitions listed below describe specific terms used:

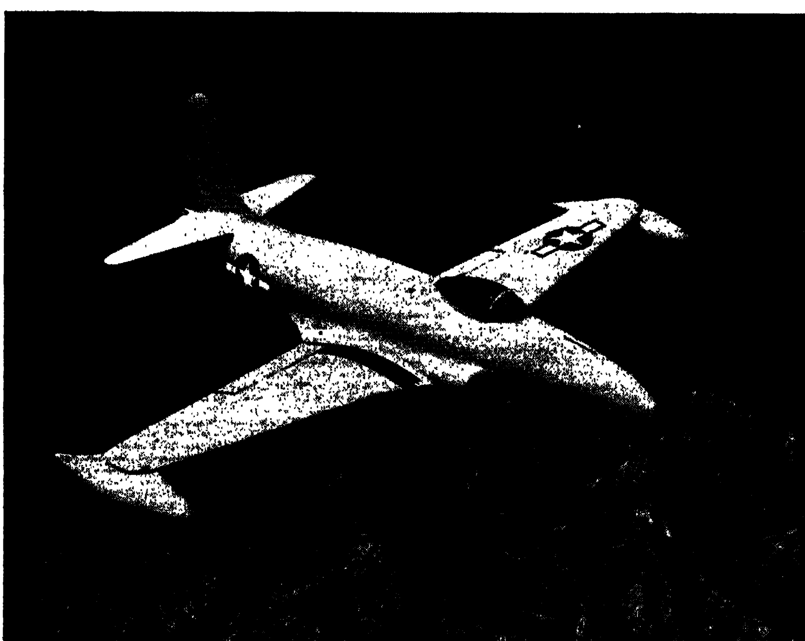
The *front* of the power plant is that part on which the accessories are mounted.

The *right and left side* are determined by looking at the unit from the rear, or exhaust-cone end.

The *rear* is that part on which the exhaust cone is mounted.

The *top* of the unit is determined by the lifting boss on the compressor diffuser in line with the adapter for combustion chamber No. 1.

The *bottom* is determined by the drain valve on the air adapter for combustion chamber No. 8.



Courtesy of Lockheed Newsbureau, Lockheed Aircraft Corporation, Burbank, Calif.

Lockheed P-80 Shooting Star.

The *direction of rotation* of the rotor is not visible from the outside because there are no visible rotating parts. Rotation is clockwise facing the unit from the front.

The *thrust* is expressed in pounds, and is the force exerted by the hot gases leaving the exhaust cone.

The *combustion chambers* are numbered 1 to 14 counterclockwise facing the center-top chamber, which is No. 1. Combustion chamber No. 8 is at the center-bottom.

The Model J-33 unit operates by the reaction principle of thermal jet propulsion. It is a further development in the series of jet-propulsion gas turbines designed and manufactured for dynamic aircraft by the General Electric Company. Like preceding models, this aircraft gas turbine is a new and unconventional aviation power plant constructed on different engineering principles and utilizing a radically different means of propulsion. Its performance on test and in the air indicates that for its weight it is the most powerful aircraft unit known to aviation.

In addition to its outstanding achievements in both speed and altitude, the J-33 unit has many important advantages which are typical only of this kind of power plant. It has, for example, the important strategic superiority of an almost instantaneous start. The comparatively long warming-up period which is characteristic of conventional airplane engines has been eliminated in the aircraft gas turbine, and the length of time required for starting has been reduced to a matter of seconds. As soon as the pilot enters the cockpit, he can begin to taxi for take-off. Because the engine has no propeller and only a few major moving parts, airplane vibration and wear and tear on the power plant and airplane have been greatly reduced and are almost negligible. The unit is much less complex in construction, light in weight, simple to operate and less expensive to maintain.

The J-33 gas turbine is in itself a complete aircraft power plant. It needs neither additional accessories, such as oil tank and coolers, superchargers, and intercoolers nor the controls necessary for the operation of such devices. Its application to

the airplane is relatively simple. The unit is suspended at three points in a specially designed air compartment in the fuselage or nacelle of the airplane. The space around the unit is so constructed that the compressor is abundantly supplied with air. In flight, this region is rammed by efficiently diffused air drawn in through an aperture at the front. Connected to the exhaust cone at the rear of the unit is a tail pipe which provides a passage of escape for the exhaust gas. In addition to the installation of the gas turbine, there are only minor connections to be made for electricity, fuel, and lubrication. An experienced crew can make a complete engine change in less than 20 min.

The turbo-jet unit has, however, certain limitations at the present time. It is essentially a high-speed power plant, operating most efficiently at its maximum output. Fuel economy is not materially improved by reducing speed since engine efficiency then falls off sharply.

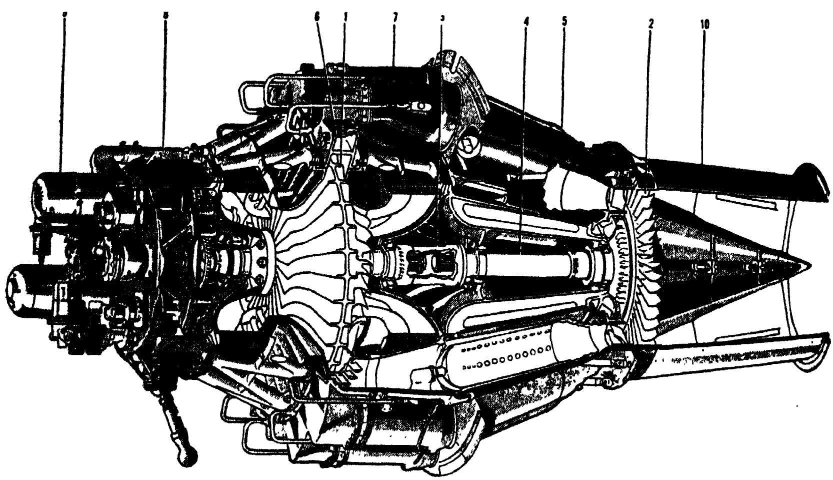
Operation of the unit is extremely simple compared to that of conventional airplane power plants. All that the pilot is required to do in order to start the engine is to throw an electric switch, push a starter button, and operate the control valve. In less than a minute, the airplane is ready for take-off. Despite the size and power of this unit, the number of instruments needed for operation is small compared to that required for the operation of other aviation power plants. Only five instruments are used; these record rotor speed, fuel pressure, lubricating-oil pressure, exhaust temperature, and turbine rear-bearing temperature. This simplicity of operation relieves the pilot of much attention to the power plant, and gives him full opportunity to devote his attention to combat or flight.

The unit is particularly adaptable to high-altitude flight inasmuch as it is both an aircraft gas turbine and a turbo-supercharger in a single unit incorporating all the speed and altitude which the alliance of these two power-producing agents makes possible. The reciprocating airplane engine loses its power as it climbs to altitude because the gradual increase in the density of the air lowers the efficiency of both the en-

gine and the propeller. Hence, in order to restore to the engine air density approximating that of sea level, it must be equipped with a supercharger or turbosupercharger to compress the rarefied air of the atmosphere for efficient operation. The propeller, however, still has to cope with the rarefied air. The J-33 gas turbine, on the other hand, climbs to altitude under its own power without requiring any additional power-increasing devices.

Description

The Model J-33 unit consists basically of a centrifugal air compressor (1) (Fig. 1) and a single-stage gas turbine (2).



Courtesy of General Electric Company

Fig. 1. Cutaway View of Power Plant.

The compressor is made up of a casing, a diffuser, and an impeller. The impeller shaft is connected to the turbine by means of the coupling (3) and the turbine shaft (4). The turbine is driven by burning a liquid hydrocarbon (kerosene) in combustion chambers (5), which receive air for supporting combustion from the compressor casing (6), through the air adapters (7). On the front of the unit is mounted the accessory-drive gear case (8) and the accessories (9). The exhaust cone (10) is mounted on the rear of the unit.

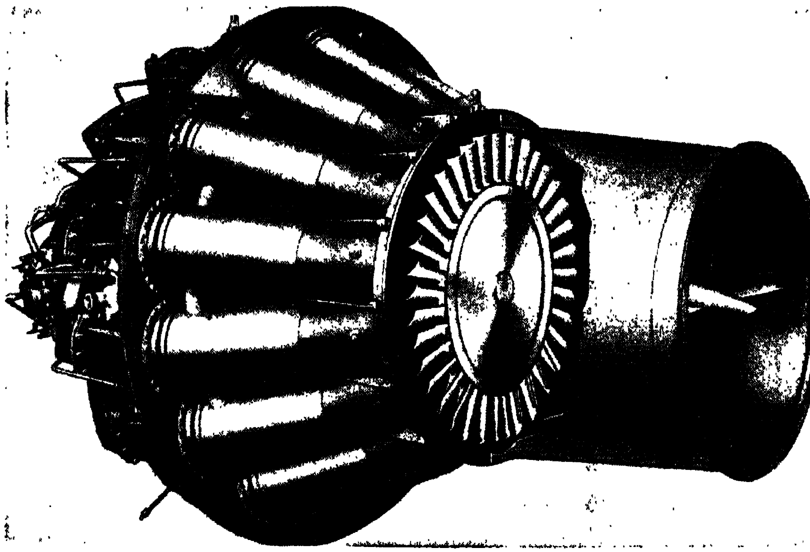
The passage of air and gas through the unit is as follows: Atmospheric air enters the compressor casing on either side and into a double-inlet multiple-vaned impeller. Compressed air leaves the impeller and enters the diffuser and compressor casing. It then passes through the air adapters to the combustion chambers. By means of a series of holes in the liners or flame tubes, air is admitted, mixed with fuel introduced by the nozzles, and burned. The exhaust gases leave the combustion chambers and pass through the diaphragm nozzles to the turbine bucket wheel. The energy of this gas passing through the turbine turns the shaft which, being connected by the coupling to the impeller shaft, furnishes the motive power for the compressor. As the gas leaves the bucket wheel, it enters the exhaust cone.

The thrust created by the gas discharged from the exhaust cone impels the airplane forward through the air by means of jet propulsion. Jet propulsion in aviation is the motive power whereby an airplane moves through space by means of gas ejected in jet form from the rear, creating the thrust of propulsion. Thrust exists when the jet velocity relative to the airplane exceeds the velocity of the airplane relative to the atmosphere. Thrust increases as jet velocity is increased and as air mass flow is increased. Both are controlled by the amount of fuel burned. The more fuel that is burned, the greater is the thrust of propulsion; therefore, the faster the airplane moves through the air. Since burning is continuous as long as the fuel flows, the airplane moves through the air in a smooth and continuous progression.

The Turbine Unit

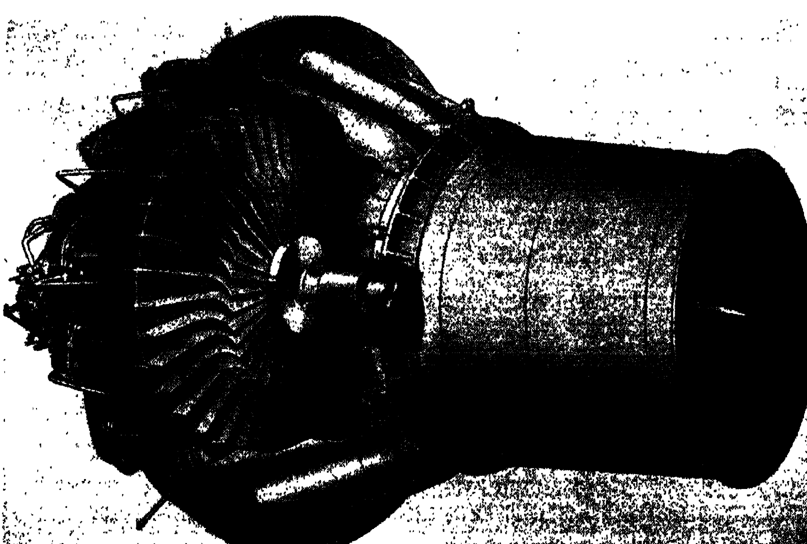
This unit consists of a turbine wheel and shaft (Fig. 2), a nozzle diaphragm, and a series of combustion chambers called the *ring-and-tube assembly*. The low-alloy steel shaft is flash-welded to the heat-resistant steel wheel. The assembly is then heat-treated to relieve strains. Securely dovetailed into the outer rim of the wheel is a continuous circle of curved blades called *buckets*.

The end of the shaft opposite the wheel is splined to fit a



Courtesy of General Electric Company

Fig. 2. Turbine Wheel in Unit.



Courtesy of General Electric Company

Fig. 3. Impeller in Unit.

coupling by means of which the turbine shaft is joined to the compressor shaft.

The nozzle diaphragm, also constructed of high-grade heat-resistant steel, is composed of two spacer rings which support a full circle of curved blades designed to direct the gas against the buckets of the wheel. A third ring, by which the diaphragm is supported in the turbine unit, has a baffle welded to it to prevent the exhaust gases from overheating the ring bearing.

The ring-and-tube assembly, in which combustion takes place for the gas turbine, consists of 14 interconnected stainless-steel cylinders which converge on a supporting ring. Each tube contains a removable liner or flame tube which is joined to its adjacent liner or flame tube by means of crossover tubes. Each tube is equipped with a piston-ring arrangement which enables the tube to slide into the air adapter to absorb the expansion caused by the heat of combustion, and to prevent air leakage.

The turbine bearings are housed in the turbine-bearing support which is located in the center of the ring-and-tube assembly.

The Compressor Unit

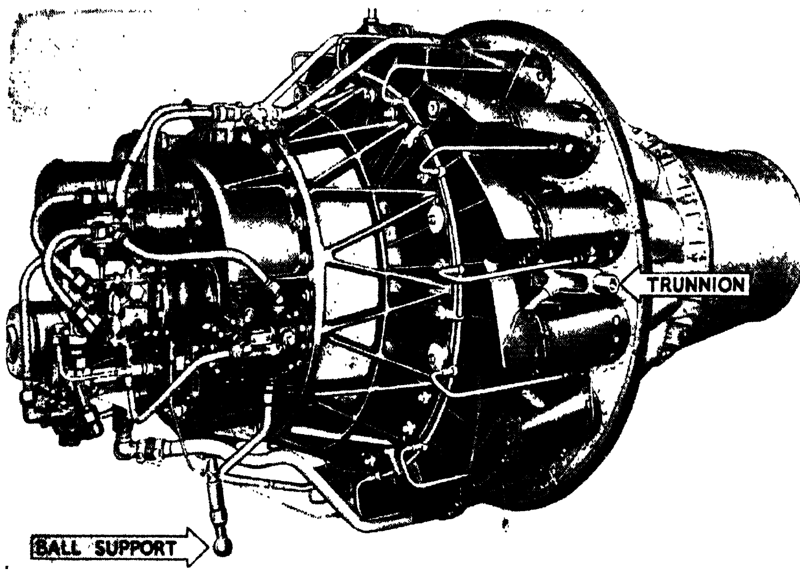
This unit is comprised of a compressor rotor, a compressor casing, a diffuser, bearing supports, guide blades, truss rings, and air adapters (Fig. 3).

The compressor rotor consists of a double-sided, multiple-vaned impeller, made from an aluminum forging, to which a stub shaft is bolted on either side. The rotor is supported by a ball bearing at the front and a roller bearing at the rear. Each bearing is housed in a bearing support.

The compressor casing consists of two half casings, made of magnesium alloy, which are separated axially by an aluminum-alloy or magnesium-alloy diffuser.

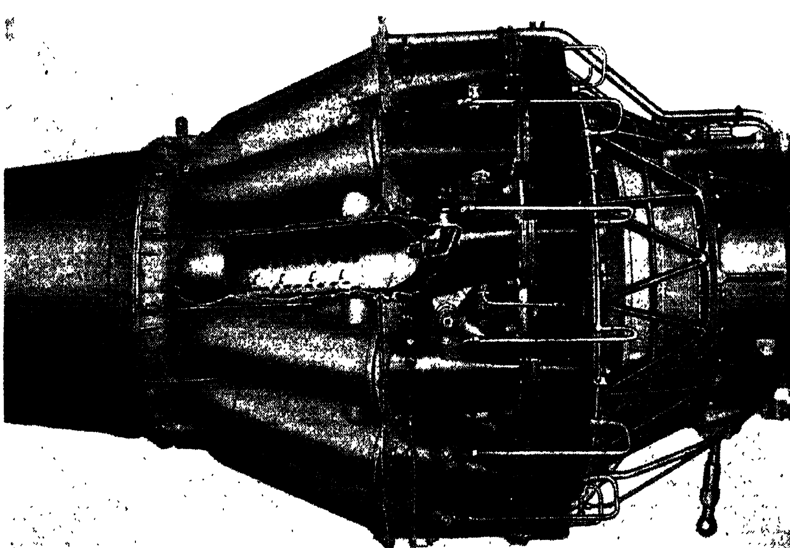
The Combustor

The diffuser is cast with 14 identically symmetrical channels radiating outwards on all sides. Through these channels,



Courtesy of General Electric Company

Fig. 4. View Showing Ball Support and Mounting Trunnion.



Courtesy of General Electric Company

Fig. 5. Cutaway View of Combustion Chamber in Unit.

the air is efficiently distributed into the air adapters leading to the combustion chambers (Fig. 4).

A truss ring is bolted to each side of the compressor casing, and serves to support the compressor bearing support. The guide blades occupy the space under the truss rings and direct the ram air into each side of the impeller. The guide blades are covered with a protective screening which prevents foreign matter from entering the compressor.

The mounting trunnions (Fig. 5) are located on the turbine side of the compressor casing in a horizontal plane. The ball support, which is the third point of suspension, is attached to a pad at the bottom or top of the gear casing.

Air Adapters

The diffuser of the compressor is connected to the ring-and-tube assembly of the turbine unit by 14 air adapters and spacers. The spacers adjoin the diffuser openings. The combustion-chamber end of each air adapter is recessed to absorb the tube expansion caused by the heat. Each air adapter contains a dome-and-nozzle assembly through which the fuel is injected into each combustion chamber.

The air adapters for outer tubes (7) and (14) each have a mounting pad for a spark plug which ignites the fuel. The air adapters (5 to 11) have outlets for draining unburned fuel through a drain manifold from the combustion unit. This unburned fuel flows by gravity from the upper chambers to chambers (5) and (11) and thence out of the unit.

Rotor Assembly

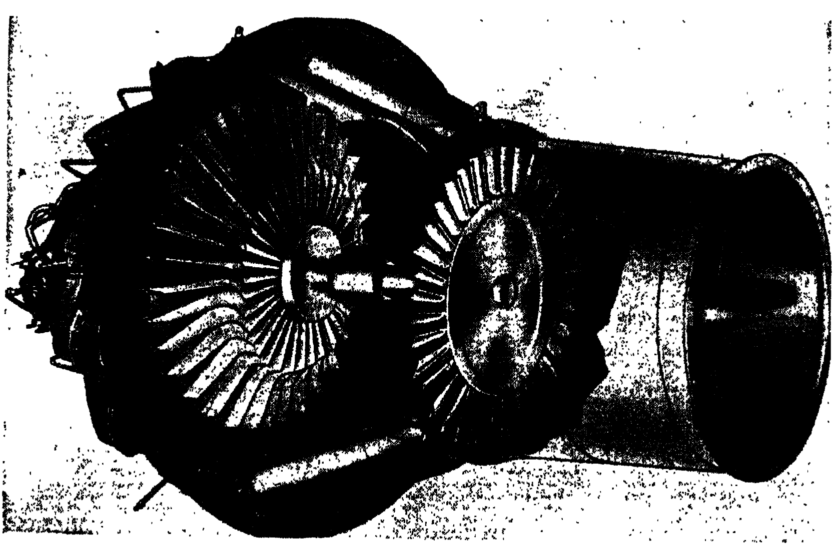
The compressor rotor and the turbine rotor are united by a sleeve-coupling assembly to form the complete rotor assembly (Fig. 6). After each rotor has been built up, the compressor rotor is checked for runout, and then both are separately balanced. During the assembling of the unit, the internally splined coupling sleeve fits over the two externally splined coupling hubs that are fastened on the end of each rotor shaft, linking the two rotors securely together and thus forming the complete rotor assembly. The complete assembly revolves in

the unit on the four high-speed antifriction bearings. The front shaft of the compressor rotor has an internal spline into which is fitted a splined coupling shaft which drives gears of the accessory-drive assembly.

The Exhaust Cone

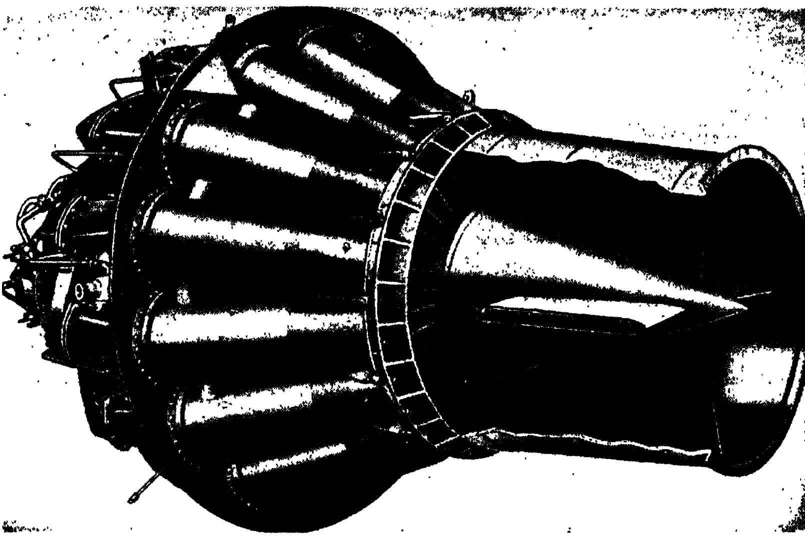
This cone is a tapered, cylinder-shaped outlet for the exhaust gas (Fig. 7). It has a closed, smaller cone within it around which the gas is ejected in a gradually expanding jet form. All of the turbulence of the expanding gas is concerted into direct thrust as it is discharged. Bolted to the turbine unit over the turbine wheel, the exhaust cone forms the rear end of the unit.

The inner cone is supported to the outer cone by four streamlined vanes called *brace assemblies*. The exhaust cone itself is made of stainless-steel sheets, reinforced at each end with stainless flanges. To keep as much of the heat energy as possible within the exhaust cone, it is covered with layers of aluminum foil separated by layers of bronze screening. The



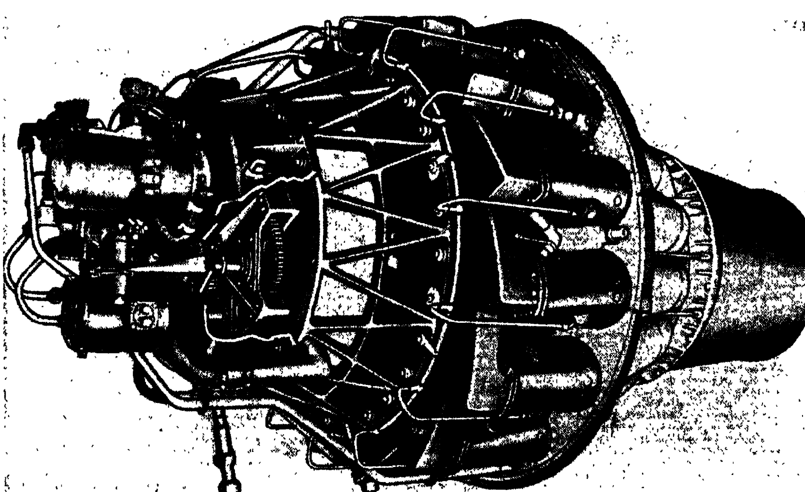
Courtesy of General Electric Company

Fig. 6. Complete Rotor Assembly in Unit.



Courtesy of General Electric Company

Fig. 7. Cutaway View of Exhaust Cone in Unit.



Courtesy of General Electric Company

Fig. 8. Cutaway View of Accessory-drive Casing Showing Gear Train.

outer layer of screening is laced together with stainless-steel lock wire.

The Accessory-drive Assembly

This assembly is mounted on the front of the unit. It consists of a gear case, a train of reduction gears, an oil reservoir, and accessories (Figs. 8 and 9).

The following accessories are mounted on the magnesium accessory case:

Generator

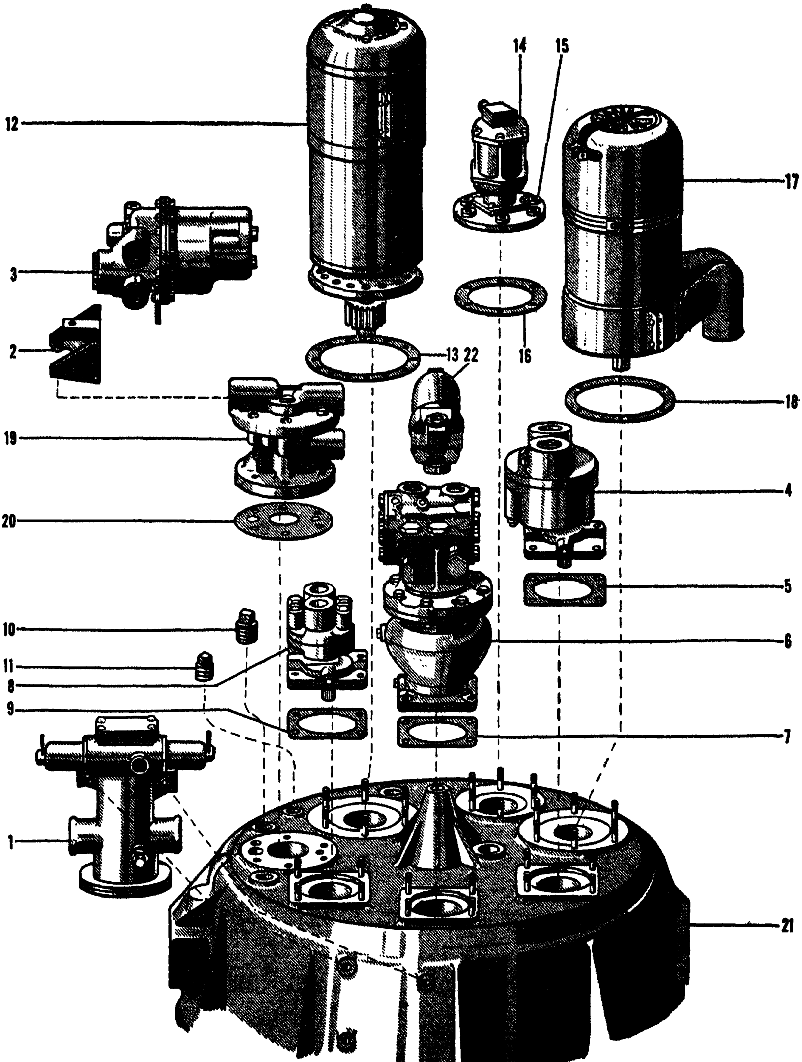
The generator is a six-pole, direct-current generator with shunt, compensating, and commutating windings. It is rated at six kilowatts, 200 amp, 30 v, at speeds varying from 4400 to 8000 rpm. The generator is mounted on the accessory gear case at the front of the unit between the tachometer generator and the main fuel pump.

Starter

The starter is a four-pole, compensated, commutating-type motor, rated at 17 v, 300 amp, 8000 rpm, for intermittent operation. It is mounted on the accessory gear case at the front of the unit between the tachometer generator and the lubricating and scavenger pump. The driving end of the starter is geared to the compressor rotor shaft through the starter gear and the overriding clutch in the gear case. The starter is used to turn the engine rotors only until the combustion of the unit is self-sustaining.

Main Fuel Pump

The main fuel pump is a pressure-loaded, gear-type pump, of positive displacement. It is mounted on the accessory gear case adjacent to the generator, and rotated clockwise as an observer looks at the front of the unit. It has a rated flow of 20 gpm at 3400 rpm and 500 psi discharge pressure. It is a single-element gear pump with constant displacement at any one speed, and has no relief valve.



Courtesy of General Electric Company

1—Control Valve
2—Bracket for Barometric
3—Barometric
4—Main Fuel Pump
5—Gasket
6—Governor
7—Gasket
8—Starting Fuel Pump
9—Gasket
10—Plug
11—Plug
12—Starter
13—Gasket
14—Tachometer Generator
15—Tachometer-generator Adapter
16—Gasket
17—Generator
18—Gasket
19—Lube and Scavenger Pump
20—Gasket
21—Accessory-drive Gear Casing
22—Lube-oil Filter

Fig. 9. Exploded View of Accessories.

Starting Fuel Pump

The starting fuel pump is a positive-displacement pump of the spur-gear type. It is mounted on the accessory gear case at the front of the unit between the governor and the lubricating and scavenger pump. Its rated flow is 4.2 gpm at 5000 rpm and 200 psi discharge pressure. Its rotation is counterclockwise as viewed from the front of the unit.

The function of the starting fuel pump, which is driven by the starter gear, is to provide additional fuel during the starting period of the engine. It is connected in parallel with the main fuel pump and has a check valve mounted on the discharge port which prevents by-passing fuel from the main fuel pump through the starting fuel pump during the regular operation when the starting pump is not running.

Lubricating and Scavenger Pump

The lubricating and scavenger pump is a mechanically driven, two-element, pressure and suction pump of the rotary type. It is mounted on the accessory gear case slightly below the center, at the front of the unit. Its lubricating element is rated to give a flow of three gpm at 2400 rpm. Its scavenging element is rated at ten gpm at 2400 rpm. The total power required to drive the pump at the above ratings is between one and two horsepower.

Governor

The governor is a by-pass valve, controlled by centrifugal flyball weights. The valve acts to prevent overspeed of the engine in excess of 11,500 rpm. At this speed, the governor rotates at approximately 3400 rpm. Centrifugal force causes a weight-and-spring assembly to expand outward and contract vertically. In contracting, the spring contacts a spring-loaded spindle which operates a linkage mechanism controlling the by-pass valves. The governor is mounted on the accessory gear case at the front of the unit between the main fuel pump and the starting fuel pump. It rotates counterclockwise as viewed from the front of the unit.

Tachometer Generator

The tachometer generator is a two-pole, three-phase, alternating-current generator which is used with an indicator to record rotor speed. It is mounted on the accessory gear case at the front of the unit between the starter and the generator. Looking at the front of the unit, its rotation is counterclockwise. The gear ratio of its gear drive to the rotor shaft is 1 to 2.404.

Fuel Control Valve

The control valve is a manually operated valve of the restrictive type which controls the amount of fuel being passed to the engine. The valve contains two elements, a stopcock and a throttle, enclosed in a single casing. These elements control the fuel supply during the starting procedure, provide speed control while the unit is in operation, and act as a final seal for the fuel system when the unit is shut down. The control valve is mounted in the accessory gear case at the front of the unit.

Oil Heater and Oil Thermostat

Provisions are furnished in the accessory gear case for two oil heaters and one oil thermostat. These units may be installed by removing threaded plugs and inserting the accessories.

The accessories described below are not mounted on the magnesium accessory case, but are attached elsewhere on the unit.

Drip Valve

The drip valve is a spring-loaded, ball-type valve. Its purpose is to drain the fuel manifold of all fuel when fuel pressure within the manifold falls below five pounds per square inch pressure. The valve is connected to the lower part of the fuel manifold in alignment with combustion chamber (8).

Combustion-chamber Drain Valve

The combustion drain valve is a spring-loaded, ball-type valve used to drain the combustion chambers of any unburned

fuel. It is located under the adapter for combustion chamber (8). The valve is connected to a drain manifold which, in turn, is connected to the seven lowest chambers (5 to 11) inclusive. The valve is designed to open and allow fuel to drain into the atmosphere when fuel pressure within the combustion chambers falls below two pounds per square inch pressure (gauge).

Spark Plugs

The spark plugs are used to ignite the fuel in the combustion chambers during starting. Two spark plugs are used on each engine; they are mounted in bosses on the air adapters. One plug is mounted on the air adapter for combustion chamber (7) and the other on the air adapter for combustion chamber (14). They are spark plugs of the porcelain-core type and have electrodes long enough to extend into the combustion chambers.

Fuel Check Valves

Spring-loaded check valves in the discharge lines of both main and starting fuel pumps prevent a reverse flow of fuel through either pump during starting and normal operation.

Lubricating-oil Filter

The lubricating-oil filter is a paper-type element which is designed to remove foreign matter from the oil feed lines. It is mounted on the discharge side of the lubricating and scavenger pump. All oil passing through the lubricating element of the pump is filtered before being passed on to the rest of the system.

Barometric Fuel Control

The barometric fuel control is a pressure-regulating valve which automatically provides the control valve with the correct amount of fuel to maintain constant speed as altitude changes. As the density of the air decreases at high altitude, a lighter load is imposed on the compressor, and, therefore, less fuel is required to maintain constant speed. During changes of altitude, it is the function of the barometric fuel control to

by-pass the surplus fuel back to the inlet line of the main fuel pump. The barometric fuel control is mounted on a bracket in the lower-center section of the accessory gear casing at the front of the unit.

Oil-pressure Transmitter

The oil-pressure transmitter is a direct-current, bellows-actuated Selsyn device which transmits to an indicator the pressure of the oil in the lubricating oil system. It is mounted on a bracket which is attached to the mounting pad of the tachometer generator on the front of the accessory gear casing. The transmitter contains a bellows which expands or contracts as pressure in the system changes. This expansion and contraction of the bellows moves a contact arm in the direct-current Selsyn device. As the arm moves, it changes the flux path of the stator in the indicator and lines up a permanent magnet attached to the indicator pointer, thus visibly showing the oil pressure. The transmitter is designed to record oil pressures of from 0 to 25 psi.

Fuel-pressure Transmitter

The fuel-pressure transmitter is a direct-current, bellows-actuated Selsyn device, rated 24 v, which transmits to an indicator the pressure of fuel in the fuel system. It is mounted on a shockproof bracket on the under side of the engine, and is connected by tubing to the drip valve. It is electrically connected to the fuel-pressure indicator in the instrument panel in the cockpit. The transmitter contains a bellows which expands or contracts as fuel pressure in the system changes. This expansion and contraction of the bellows moves the contact arm in the direct-current Selsyn device. As the arm moves, it changes the flux path of the stator in the indicator and lines up a permanent magnet attached to the indicator pointer, thus visibly showing the fuel pressure. The transmitter is designed to record fuel pressures of from 0 to 600 psi.

Ignition Coils

A continuous spark is provided at the engine spark plugs during starting by two ignition booster coils mounted on the

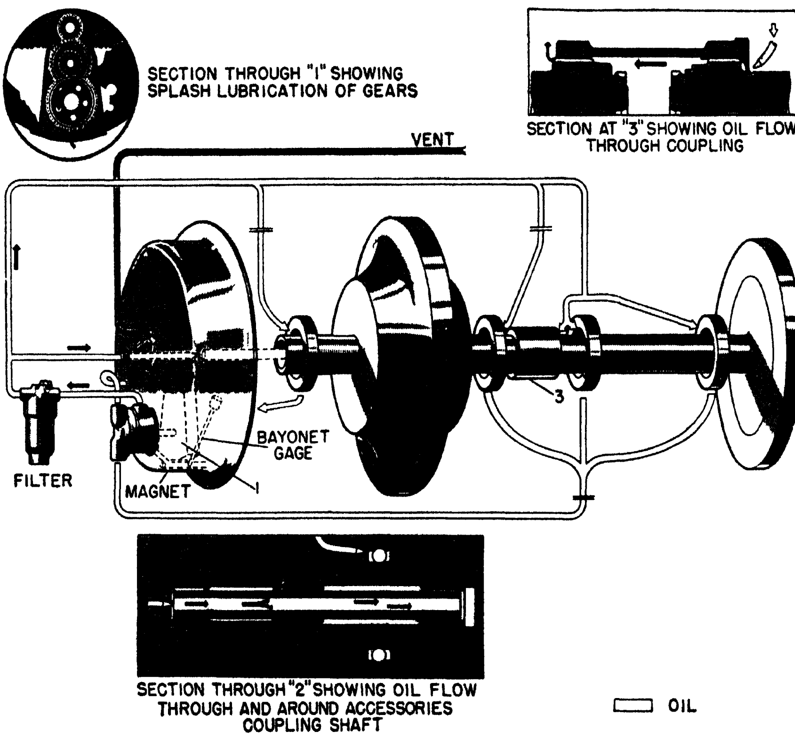
engine and wired in parallel. The coils receive power from the engine starter relay through the single-pole, double-throw, ignition booster switch in the cockpit. Operating the starter switch to start the engine automatically operates the booster coils.

Lubrication System

The unit is lubricated by a simple wet-sump system (Fig. 10). The supply reservoir is an integral part of the accessory-drive gear casing and is formed by the gear casing and the front-bearing support casing. The maximum capacity of the reservoir is 14 qt. The minimum supply allowable is 6 qts. Measurement of the oil may be determined with a bayonet gauge which is attached to the filler cover located on the left side of the accessory gear casing when facing the front. The lubricating system is adequate for 10 hr of continuous operation at normal rated speed.

Lubrication oil is delivered under pressure to the bearings by means of a two-element lubricating and scavenger pump. The pump is located on the accessory gear casing, and has a port on its mounting flange which is in direct contact with the oil in the reservoir. The lubricating oil passes from the lubricating element of the pump to an oil filter, and through external tubing to the four rotor bearings and the couplings. The oil is directed into the coupling and bearings through jets with $\frac{1}{16}$ in. diam orifices, and into the large gear coupling through a jet which has a $\frac{1}{8}$ in. diam orifice. Oil seals are provided to prevent oil leakage past the front and rear compressor bearings and the rear turbine bearing. Unbalanced pressures between the inside and outside of the casing that contains the bearings provide a further check to oil leakage. Oil is returned to the reservoir by gravity and suction. The front compressor bearing and the quill shaft of the accessory drive both drain directly into the reservoir. The oil from the three other bearings and from the coupling drain into a sump; from the sump it is drawn by the scavenger element of the lubricating and scavenger pump and returned to the oil reservoir.

All of the gears and bearings in the accessory-drive casing are lubricated by means of a splash system emanating from the gear which drives the lubricating and scavenger pump. This gear is located in a special compartment under the oil level. Oil is admitted to the compartment through an orifice which controls the quantity of oil that is used for lubricating



Courtesy of General Electric Company

Fig. 10. Diagram of Lubricating System.

the gears and bearings, and prevents the oil in the reservoir from being churned into foam. A baffle in the reservoir prevents surging of the oil during maneuvers and negative accelerations. The lubricating and scavenger pump is so located that its inlet is supplied with oil at minimum oil level in a zero-deg diving angle with a 10-deg inclination to either side, or a zero- to sixty-deg climbing angle with a 10-deg inclination to either side. It is possible to maintain inverted

flight positions for short periods of one minute or less without an appreciable loss of oil. The lubricating pump runs dry under inverted flight conditions, and the spline and gear couplings will operate for short periods without being damaged. A vent tube, which passes over the gear case rearward to the air baffle, vents the reservoir to the aft section of the airplane nacelle. Oil pressure at all times may be observed on an indicator located on the control panel of the airplane.

Fuel System

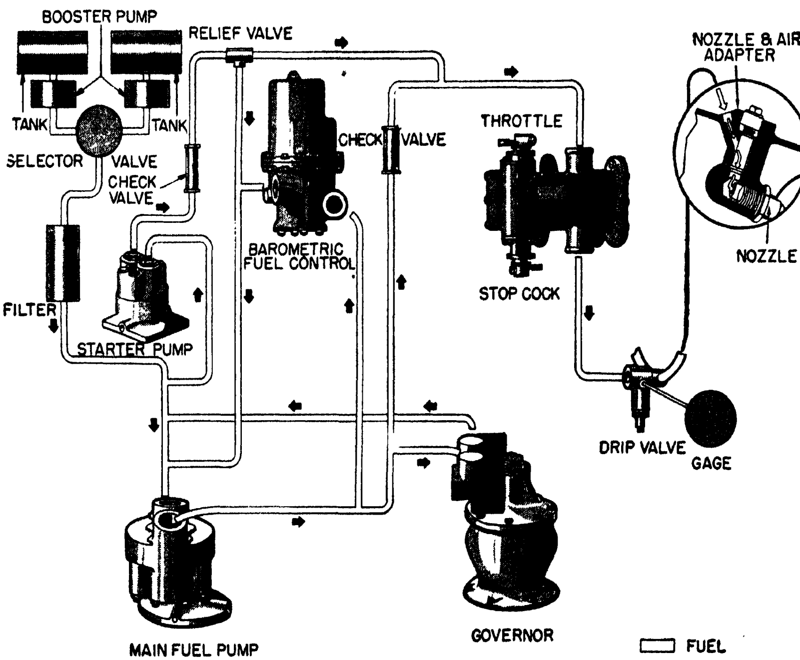
The fuel system contains the following elements (Fig. 11):

Airplane fuel tank
Booster pump
Selector valves
Two fuel filters
Main fuel pump
Starting fuel pump and check valve
Barometric
Governor
Control valve (throttle and stopcock)
Drip valve
Drain manifold and drain valve
Fuel nozzles

When the rotor is actuated by the starter, the main fuel pump is assisted by the starting fuel pump in building up fuel pressure and delivering fuel in sufficient quantities to the nozzles for starting. After the rotor comes up to approximately 1700 rpm, the main fuel pump takes over and delivers fuel to the nozzles for further increases in speed; at this time, the check valve isolates the starting fuel pump, and it is automatically shut down. Both the governor and the barometric act as a by-pass to the fuel pump, as this pump has constant displacement characteristics. The governor by-passes fuel in order to prevent speed exceeding 11,000 rpm, which is the maximum allowable speed. The barometric by-passes fuel in order to maintain, for a given cockpit setting of the control valve, a constant speed regardless of the altitude. By means of the control-valve throttle, the pilot can vary the speed of the gas turbine by changing the amount of the fuel flow to the nozzles. The stopcock side of the control valve is used to shut off com-

pletely the flow of fuel. A drain manifold and drain valve, which are connected to the lower combustion chambers, drain the fuel in order to avoid accumulation in the combustion chambers when the unit is shut down. Each nozzle in the 14 combustion chambers contains a metering device to provide equal distribution of fuel in all the chambers.

In the event of the main-fuel pump failure, the starting



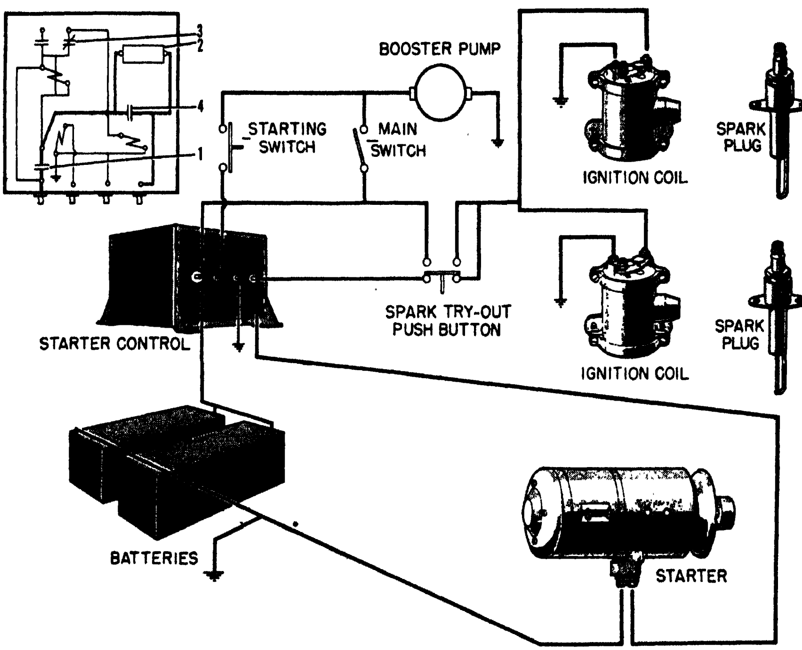
Courtesy of General Electric Company

Fig. 11. Diagram of Fuel System.

system may be used to operate the unit at reduced rating. This is accomplished by keeping the starter button manually depressed during the time that the emergency fuel is required. Operation of the starting fuel pump alone will give, at sea level, about one fourth of the maximum thrust which can be obtained. At sea-level static conditions, this thrust is about 1000 lb at 7000 rpm rotor speed. At altitude conditions, the percentage of maximum thrust is much greater.

Ignition System

Referring to the schematic ignition diagram (Fig. 12), the main switch is closed, furnishing power to the booster pump and energizing one side of the starting switch. The booster pump furnishes fuel pressure to the starting fuel pump and



Courtesy of General Electric Company

Fig. 12. Schematic Ignition Diagram.

to the main fuel pump, both of which are mechanically driven by the rotor.

Closing the starting switch energizes the relay in the starting control, and thereby closes contactor (1) which completes the power circuit from the batteries to the starter through the starting resistor (2). This starting resistor limits the starting current and turns the motor over gently to *take up* the clutch without shock. Three quarters of a second after the closing of contactor (1), the contacts of the time-delay relay (3) close, thus energizing and closing contactor (4).

Closing of contactor (4) short-circuits the starting resistor (2). This allows the full battery voltage to be applied to the starter, thus rapidly accelerating the unit to its starting speed of 1000 rpm in approximately 15 sec with fully charged batteries. When contactor (1) closes, it closes the circuit to the spark plug; hence, ignition is present immediately upon the actuating of the starter.

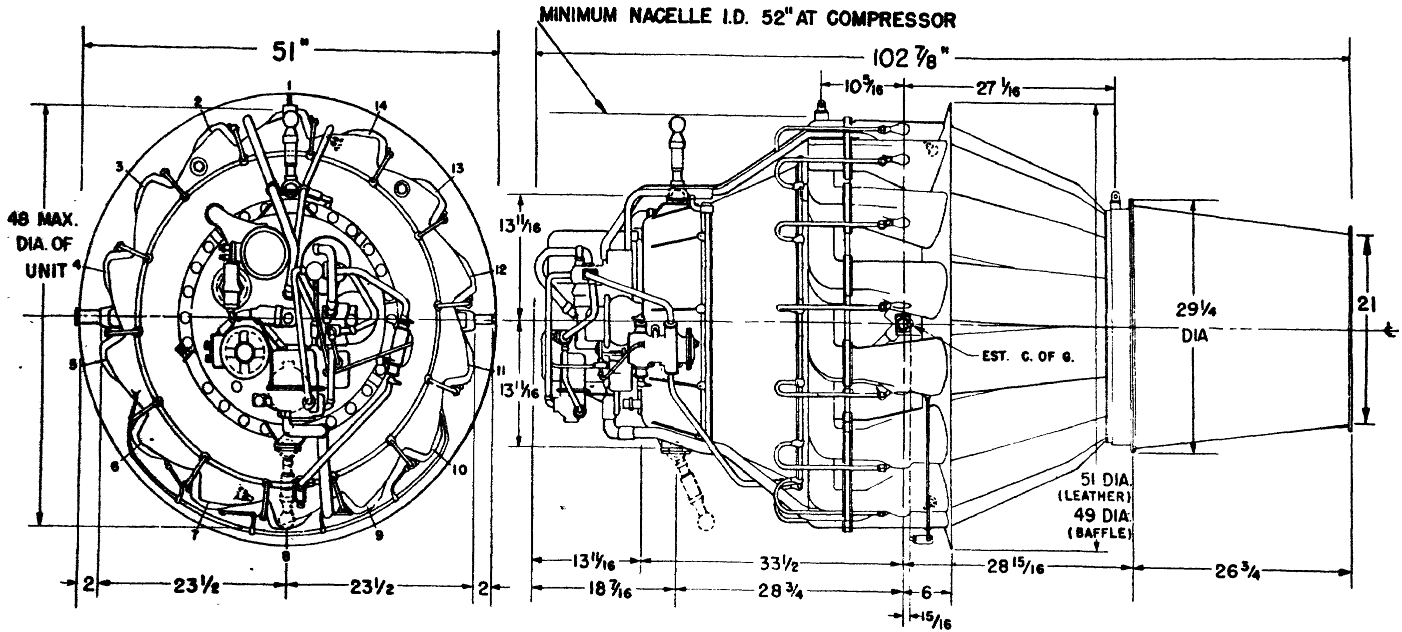
When the rotor reaches 1000 rpm, fuel pressure is built up, and fuel is injected into the combustion chambers and ignited by the spark plugs. At approximately 1700 rpm, the clutch automatically disengages and sets the starter and starting fuel pump free from the rotor. At this time the operation of the unit is self-sustaining. When the gas turbine has thus taken over and is running on injected fuel, the pilot removes his finger from the starting switch to relieve the starter of further work.

Cooling System

The J-33 gas turbine is air-cooled. The cooling process requires a maximum output of approximately two per cent of the air flow through the unit. A cooling air fan on the front side of the turbine wheel maintains cooling air flow independent of the ram pressure available in flight. Cooling air is directed over the rear bearing housings, the turbine wheel, and the combustion chambers before being discharged from the system.

Gas Turbine Mount

The unit is mounted in an airplane by means of two side trunnions and a front ball support (Fig. 4). One of the trunnion spindles is designed to slide in its mounting to accommodate expansion and contraction of the engine. The front ball support, which may be attached either to the top or the bottom of the accessory case, depending on installation requirements, is adjustable to permit correct alignment of the unit in the airplane.



Courtesy of General Electric Company

Fig. 13. Outline of Unit.

The pages (303-313) omitted from Chapter XV dealt with detailed maintenance of the jet unit under discussion and were considered sufficiently confidential to be omitted at the request of the Army Air Force censors. It is hoped that this information will be available for publication in a later printing of the book. The author is pleased to be able to publish as much of Chapter XV as the censors would allow at this time, as it is the author's object to give the reader the latest information possible within security regulations.

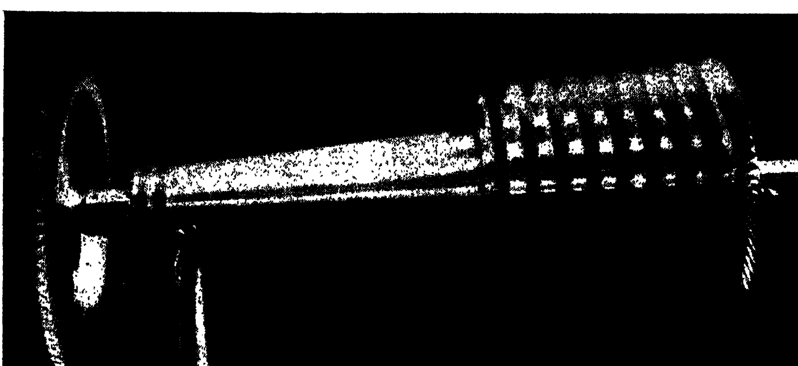
R. T. S.

Section B—British

Chapter XVI

British Aircraft Gas Turbines [1]

Current British achievement with the gas turbine is the result of two distinct factors. First, success with axial-compressors whose origin is attributed to Dr. A. A. Griffith (Fig. 1). The second factor is associated with the centrifugal com-



Courtesy of Flight *Magazine*

Fig. 1. Rotor of Metropolitan-Vickers F.2 Engine, Which Has a Nine-stage Axial-flow Compressor, a Single, Annular Combustion Chamber, and a Two-stage Turbine, All on the Same Axis. Taken by permission from *Gas Turbines and Jet Propulsion for Aircraft*, published by Aircraft Books, Inc., New York.

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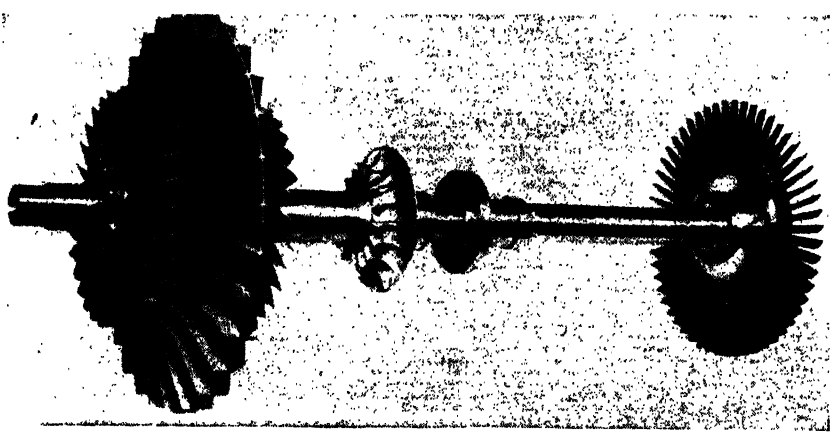
[1] *Flight,* December 20 and 27, 1945. An abstract of the 1945 Wright Brothers lecture delivered on December 17 before the Institute of the Aeronautical Sciences (the American equivalent to the British Royal Aeronautical Society) by Dr. H. Roxbee Cox, chairman and managing director of Power Jets (Research and Development), Ltd.

pressor and with the name of Air Commodore Whittle (Fig. 2). In 1926, Dr. Griffith evolved, at the Royal Aircraft Establishment (R.A.E.), an aerodynamic theory of turbine design; in 1929, he discussed the prospects of the internal-combustion turbine driving a propeller. In 1936, the R.A.E. obtained authority to build an axial-flow compressor. This compressor was damaged by an enemy bomb in 1940, but in the meantime, a more ambitious scheme had taken shape. In 1937, Mr. H. Constant of the R.A.E. suggested that it was possible to construct



Courtesy of Flight *Magazine*

Fig. 2. The Rotor of a Derwent Engine Comprises a Double-sided Impeller at One End of the Shaft, the Turbine at the Other, and a Fan for Cooling the Bearing in the Center.

a turbine to drive a propeller which would compare favorably in specific weight and fuel consumption with the piston engine. The results of the proposal made by Whittle are known. As for the suggestion made by Constant, the R.A.E. collaborated with Metropolitan Vickers and, as a result of considerable work in design, there was developed the B.10 with a nine-stage axial-compressor driven by a four-stage turbine. The B.10 was followed by a series of models in which the air flowed without bends through a single compressor, an annular combustion chamber, the compressor turbine, and a power turbine. In the meantime, an experimental turbocompressor very much like that suggested by Griffith was designed by the

R.A.E. in 1938, manufactured by Armstrong-Siddeley in 1939, and tested in 1940.

British Thompson Houston (B.T.H.), Power Jets, the Rover

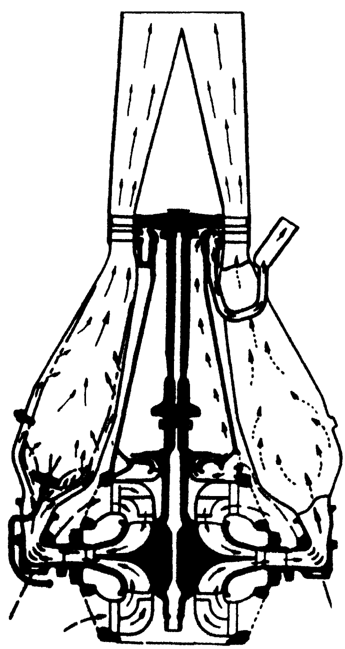
Courtesy of Flight *Magazine*

Fig. 3. The Rotor of the Halford H.1 Engine, Which Used a Single-sided Impeller and a Straight-through Combustion System.

Company, Halford-de Havilland, and Rolls-Royce co-operated in the production of these models. The Rover Company initiated the change-over to straight-through combustion. The straight-through construction, originally demonstrated on the Halford H. 1, Figures 3 and 4, is more efficient than the return-flow system of the classic Whittle engine (Fig. 5).

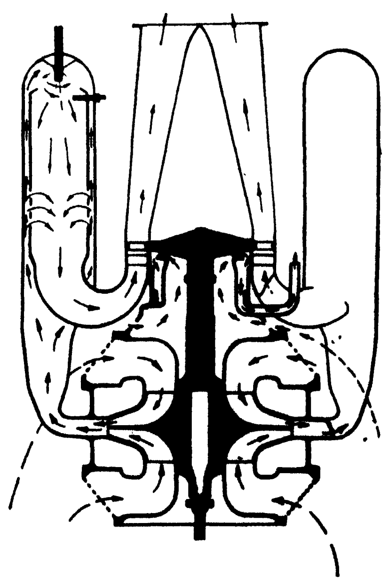
Collaborators

At the time Armstrong-Siddeley was given its first contract (November 1942), there existed an overwhelming concentra-



Courtesy of Flight *Magazine*

Fig. 4. Diagram of the Straight-through Combustion Chamber.



Courtesy of Flight *Magazine*

Fig. 5. Diagram of the Return-flow Combustion System of the Classic Whittle Engine.

Taken by permission from *Gas Turbines and Jet Propulsion for Aircraft*, published by Aircraft Books, Inc., New York.

tion on the centrifugal-compressor type of engine. Only Metropolitan-Vickers favored the axial-compressor. Armstrong-Siddeley at that time had only begun work on the A.S.X., a fourteen-stage axial-flow compressor engine having a two-

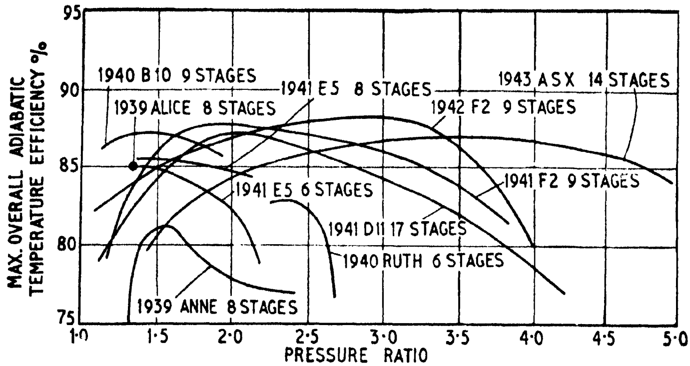


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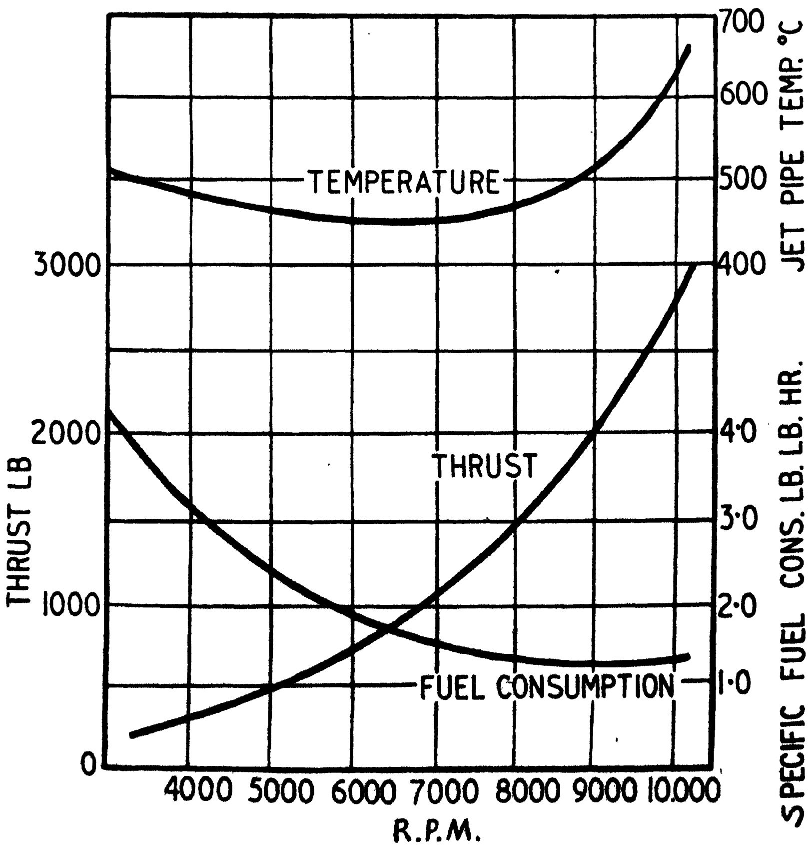
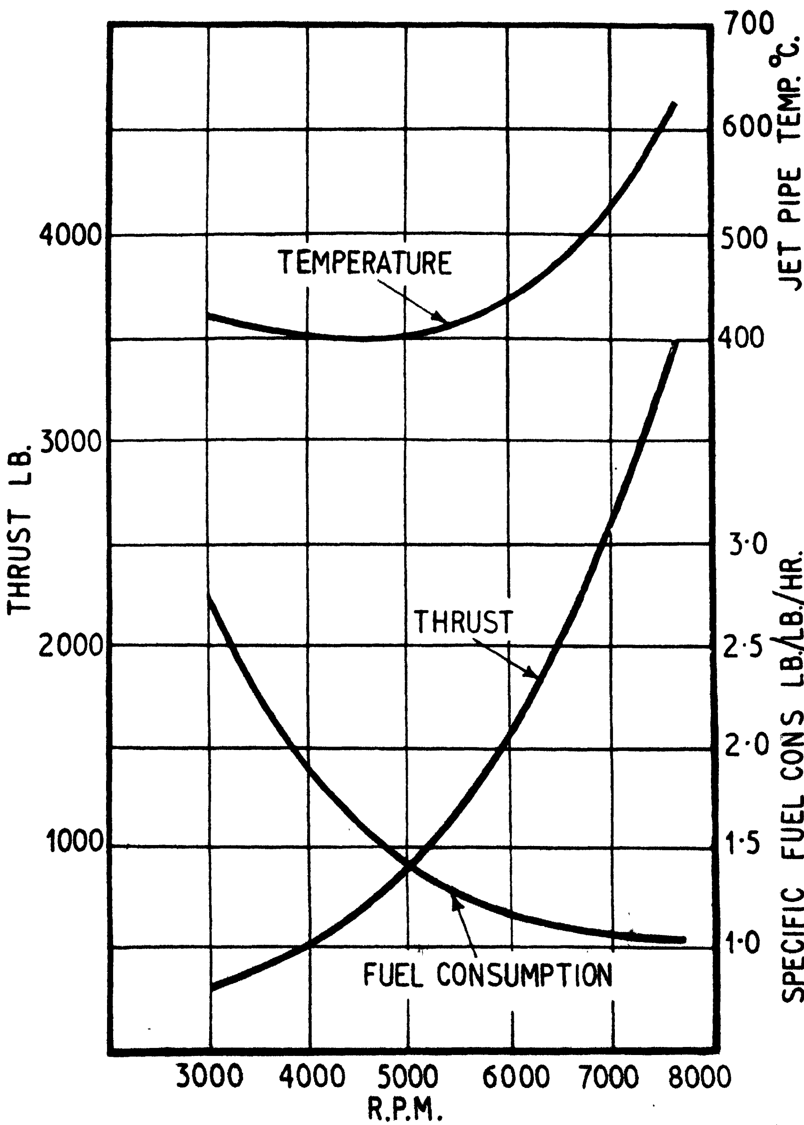
Fig. 6. Some Data on Early Axial-compressor Efficiencies.



Courtesy of Flight *Magazine*

Fig. 7. Sea-level Static Performance of the de Havilland Goblin II Production Engine.

stage turbine and 11 separate combustion chambers. Testing began in April 1943, and the engine is now giving a maximum thrust of 2800 lb in the static condition at sea level. This model is being tested in the Lancashire Universal Test Bed.



Courtesy of Flight *Magazine*

Fig. 8. Sea-level Static Performance of the Metropolitan-Vickers F.2 Engine.

Tests of the compressor on the Northampton test plant established an adiabatic efficiency of 87 per cent.

The Gas Turbine Collaboration Committee co-ordinated collaborating activities. Research teams comprised of personnel from Mond Nickel, Firth-Vickers, Jessops, and others

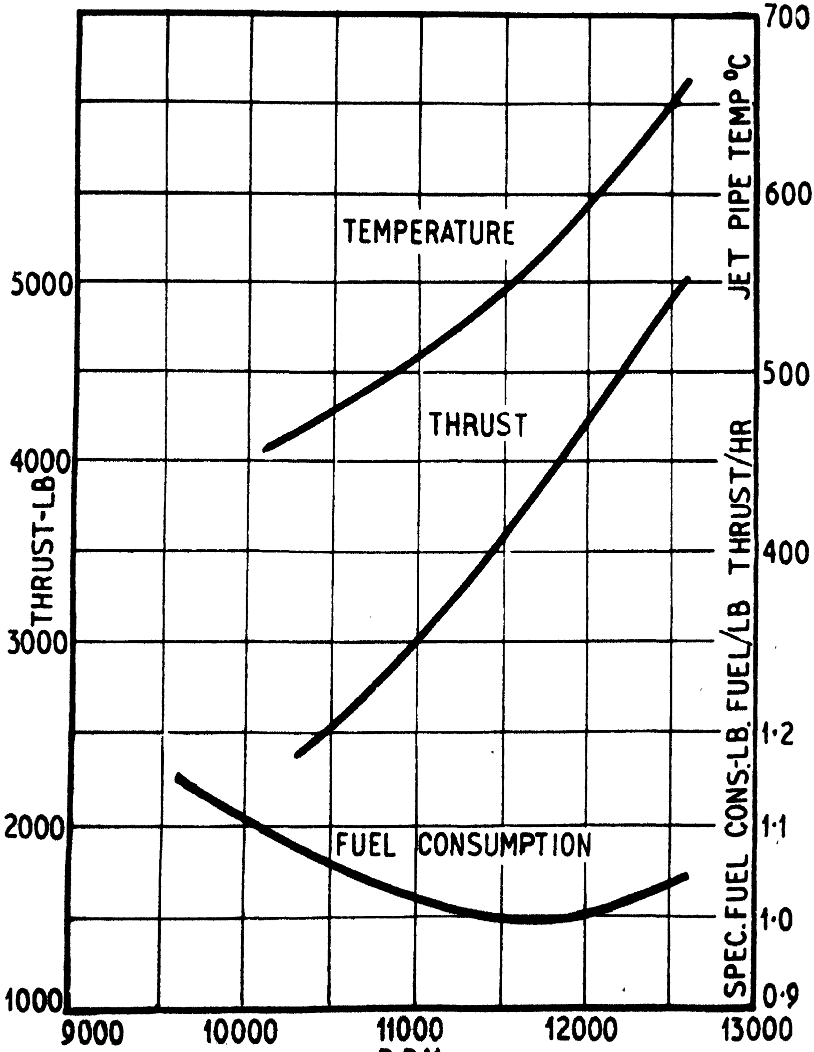


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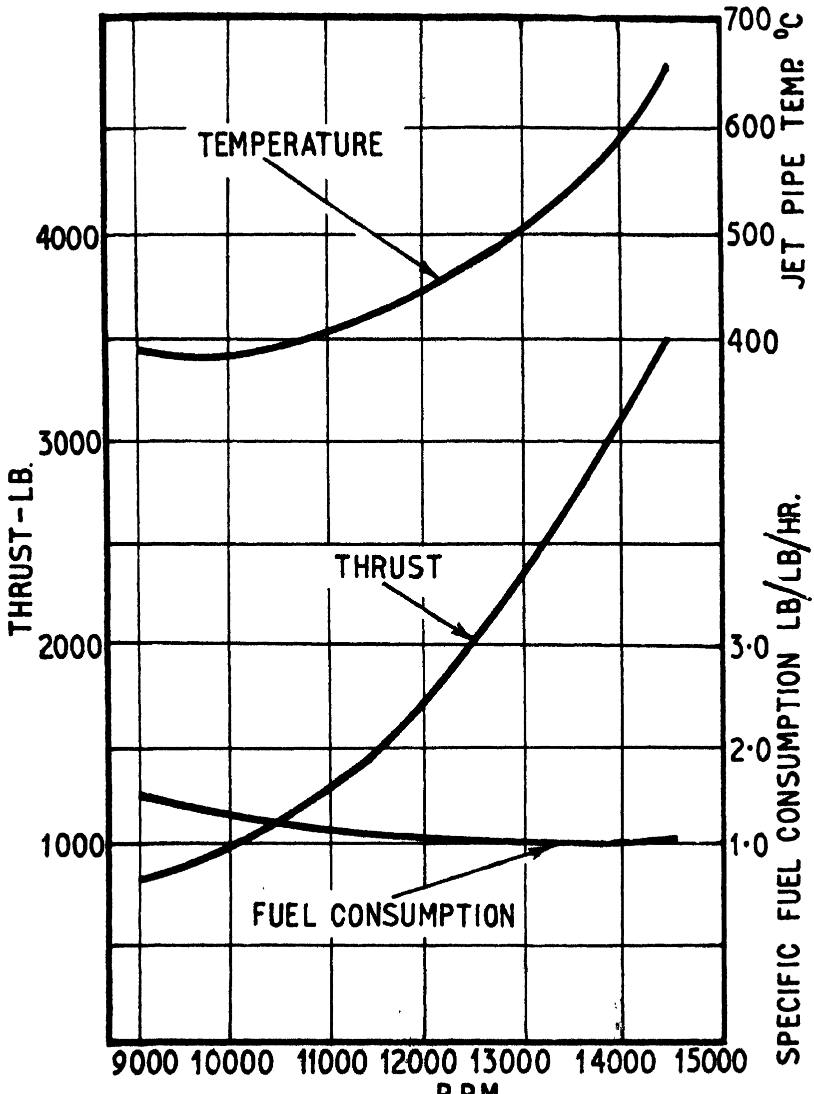
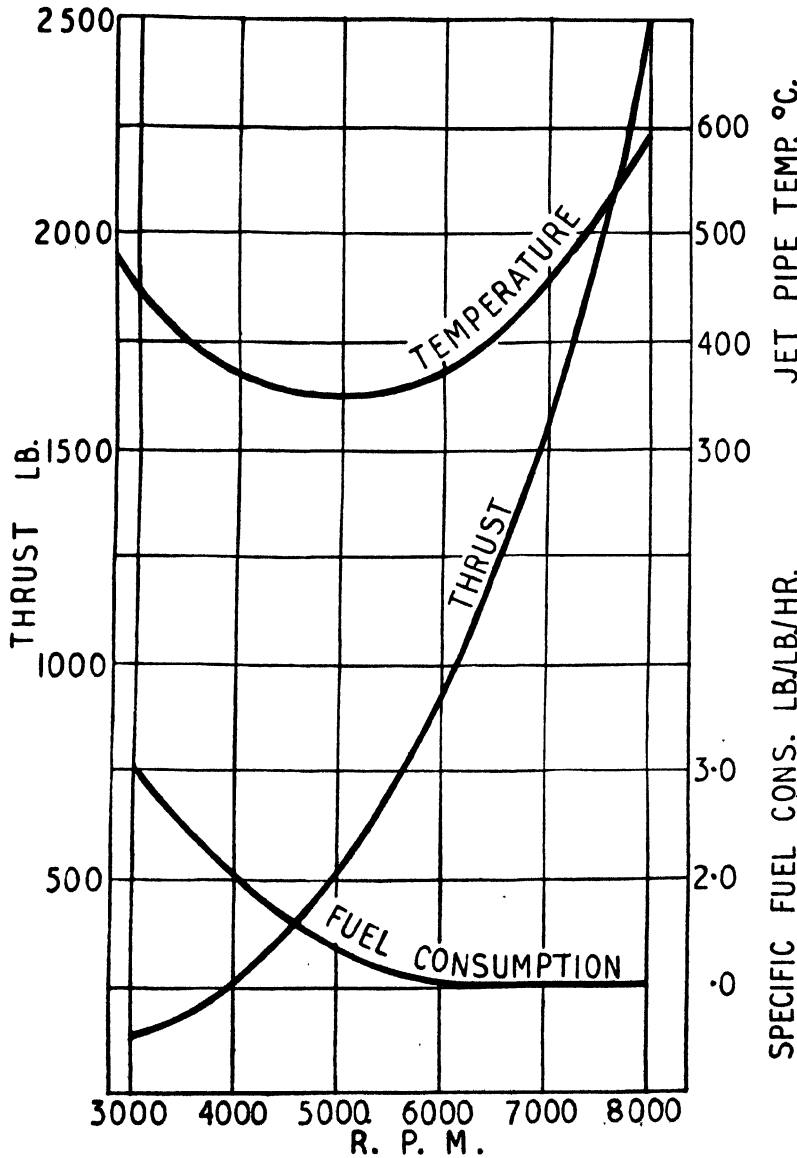
Fig. 9. Sea-level Static Performance of Rolls-Royce Nene Engine.



Courtesy of Flight *Magazine*

Fig. 10. Sea-level Static Performance of Rolls-Royce Derwent V Engine. Taken by permission from *Gas Turbines and Jet Propulsion for Aircraft,* published by Aircraft Books, Inc., New York.



Courtesy of Flight *Magazine*

Fig. 11. Sea-level Static Performance of the Armstrong Siddeley A.S.X. Engine.

worked on high-temperature materials. A team from Power Jets tackled the problem of burners. Teams from Power Jets (Research and Development), Lucas, and de Havilland co-operated in work on combustion technique. The work of the Asiatic Petroleum Company and the Combustion Panel was on the physics of combustion. Constant's team at the R.A.E. devoted itself to gas analysis, blade vibration, and blading design.

The organizational hub of British collaboration on turbines is the Ministry of Aircraft Production. Their over-all policy is determined and implemented by contract. Grouped around the Ministry and the government-owned Power Jets (Research and Development), Ltd., which conducts investigations on the design of jet units and their component parts are the Rolls-Royce, De Havilland, Bristol, Armstrong Siddeley, Metropolitan-Vickers, and British Thompson Houston companies which develop jet units of all types. All these companies are doing gas turbine work under government contract.

For the longest ranges required, the prop-jet combination is superior to the jet-propulsion gas turbine. In the long-range sphere, the competition may be between the prop-jet and the ducted-fan turbine engines of Chapter 21 and others of this type, but when comparing the prop-jet aircraft with the ducted-fan turbine aircraft, the slip-stream drag of the former may outweigh the advantages derived from the better fuel consumption of its engines. If pusher propeller turbine units are advanced to clinch the argument in favor of the propeller model, then discussion becomes complex because mass balance of the aircraft, installation problems (jets fouling propellers) and engine design (propeller shaft and exhaust at the same end of the engine) are rapidly introduced. Nevertheless, whatever the result of the competition between the two types, it would be wrong to assume that either propeller or ducted-fan gas turbines are essentially long-range engines. For comparative performance data, see Figures 6 to 11.

Chapter XVII

Bristol Gas Turbine Propeller Unit Theseus I[1]

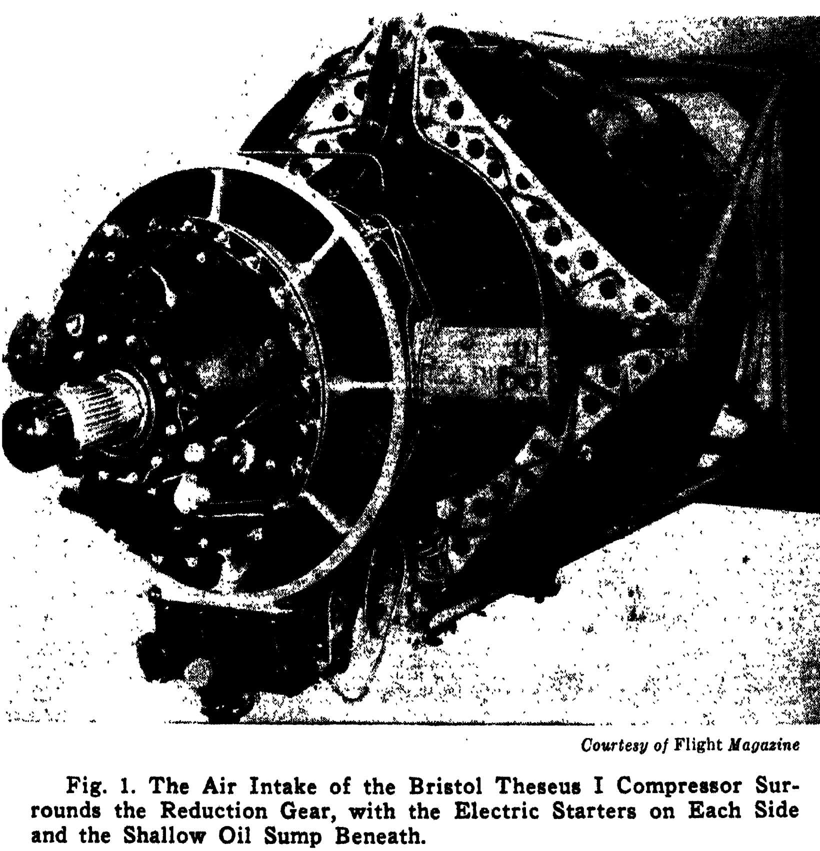
The Theseus I consists of a compound, multistage compressor which takes in air through an annular orifice around the propeller reduction gear. This combined axial and centrifugal compressor is similar to Whittle's original patent taken out in 1930. The compressed air is then delivered through a heat exchanger before passing to the combustion chamber, where its temperature is further raised by the burning of injected fuel (Figs. 1 and 2). The resulting products of combustion pass to a turbine by means of which the compressor and auxiliaries are driven, and thence to a further turbine stage which, through a forward extension shaft, drives the propeller as shown in Figure 3. After leaving the latter turbine, the gases pass through the heat exchanger, where they give up a measure of their heat to the compressed air that is on its way to the combustion chambers. From the heat exchanger, the exhaust gases are finally discharged through a controllable nozzle, thus providing a certain proportion of the total thrust.

A feature of all turbines is that they have the inherent quality of optimum efficiency at maximum speed. Consequently, when driving a propeller by means of a gas turbine is being considered, the very real problem of obtaining flexibility presents itself. A turbine will, nevertheless, provide a good proportion of its optimum power when it is run within a fairly small range of its optimum speed. It is by taking advantage of this quality that the Bristol turbine designers have done such a good job. The solution lies partly in providing separate

[1] *Flight*, Dec. 6, 1945.

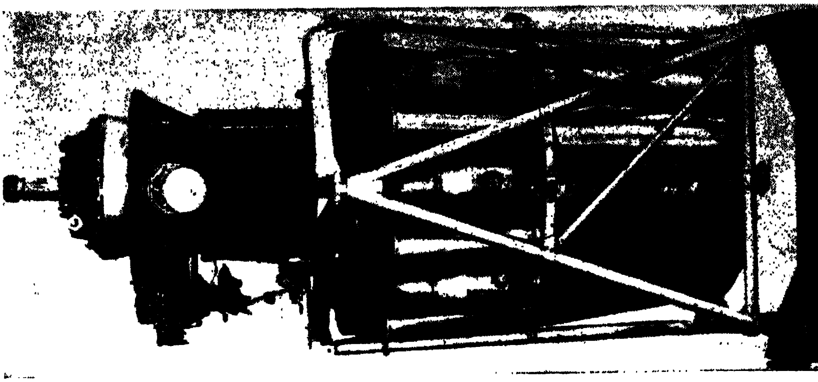
drives for compressor and propeller, and partly in an extremely clever control device. The net result is that the speed of the independent propeller drive can be varied over a considerable range without affecting the over-all turbine efficiency.

With large aircraft, it is generally more useful to increase propeller rpm rather than horsepower for take-off, and the possibility of independent choice of rpm is, therefore, a valuable feature, the usefulness of which will, no doubt, be enhanced to reduce propeller tip noise by the reduction of rpm when cruising. Curiously, the operating characteristics of the prop-jet combination (and also of pure jets) are virtually the direct opposite of those of the piston engine because, whereas with the latter, fuel consumption increases with throttle open-



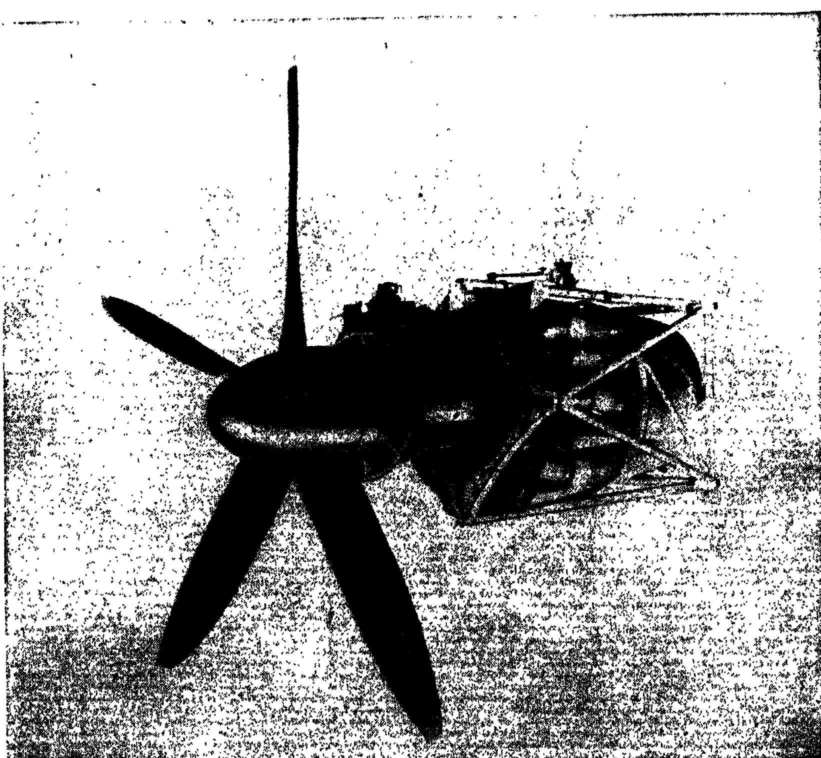
Courtesy of Flight *Magazine*

Fig. 1. The Air Intake of the Bristol Theseus I Compressor Surrounds the Reduction Gear, with the Electric Starters on Each Side and the Shallow Oil Sump Beneath.



Courtesy of Flight *Magazine*

Fig. 2. From the Compressor, Air Is Ducted Through the Parallel Pipes to the Heat Exchanger, Thence Forward to the Combustion Chambers.



Courtesy of Flight *Magazine*

Fig. 3. Fitted with a Rotol Five-blade Airscrew and Complete with Mounting Structure, the Bristol Theseus I Makes a Neat Installation.

ing and height, in the case of gas turbines, fuel consumption goes down with increase of speed and height. Thus, one has the seeming anomaly that, if the engine is kept at full power, consumption goes down with gain in height, but consumption per hp increases when the throttle is closed during level flight.

The designers on the Bristol project were given the job of designing a turbine for long-range aircraft; the objective set them at that time was to produce a turbine which would have an over-all fuel consumption comparable with that of a piston engine at 300 mph at 20,000 ft. This amounts to 0.57 lb per bhp-hr at sea level, 0.51 at 20,000 ft altitude, and 0.48 at 40,000 ft altitude.

These conditions were postulated as the lowest speed and altitude at which the turbine could compete with conventional engines; above these speeds and altitudes, turbine efficiency increases. The Bristol Aeroplane Company did not believe that the values could be less than these, and 20,000 ft was chosen because pressure cabins were thought inevitable, and these cabins would not be worth while at lower altitudes.

Since gas turbines have no cooling system other than a small oil cooler for the reduction gear, it was concluded that, if the designers could get down to within 5 to 10 per cent above piston-engine consumption, they could successfully compete with such engines. This was fixed as the full load condition, since turbine performance is most efficient at full power and it was not necessary to provide for any other condition except that of top speed.

Chapter XVIII

De Havilland Goblin II[1]

When the de Havilland design was first contemplated in 1941, one of the most important considerations was the time element. A turbine jet engine was needed quickly, and anything likely to prolong its development or unduly to complicate the manufacture had to be avoided. Under different circumstances, an axial-flow compressor and a single annular combustion chamber might have been developed for the first de Havilland unit. However, although both of the latter were theoretically superior to the centrifugal compressor and individual combustion chamber, practical data concerning them were lacking and, consequently, they were as yet unknown quantities.

One of the principal differences between the Goblin series of Figures 1 and 2 and other units is in the use of a single-sided impeller for the centrifugal compressor, similar to Figure 3, Chapter 16. Although a double-sided impeller, as was favored by Whittle, would result in a unit having a smaller overall diameter and although a large-diameter impeller (in addition to being more limited in rotational speed) would raise some problems in manufacture, the de Havilland design staff believed that the preponderance of evidence favored the single-sided design.

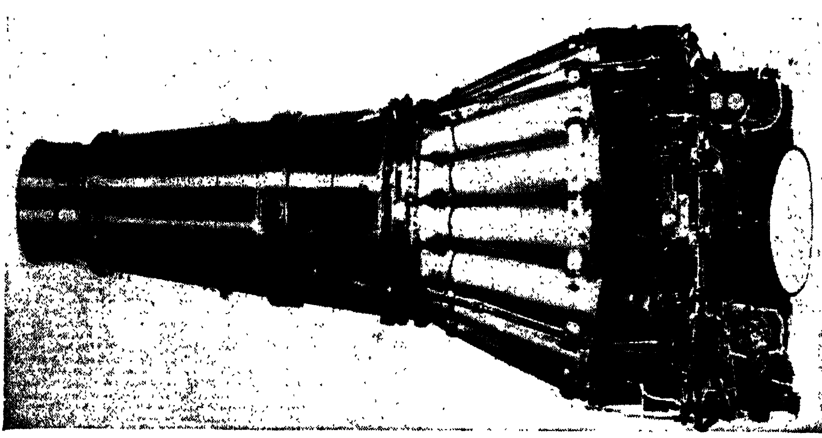
Direct Air Intake

For a unit with a single-sided impeller, two more advantages are claimed. First, that inherent simplicity and compactness permit the use of only two bearings for the main rotating

[1] *Flight,* Feb. 21, 1946.

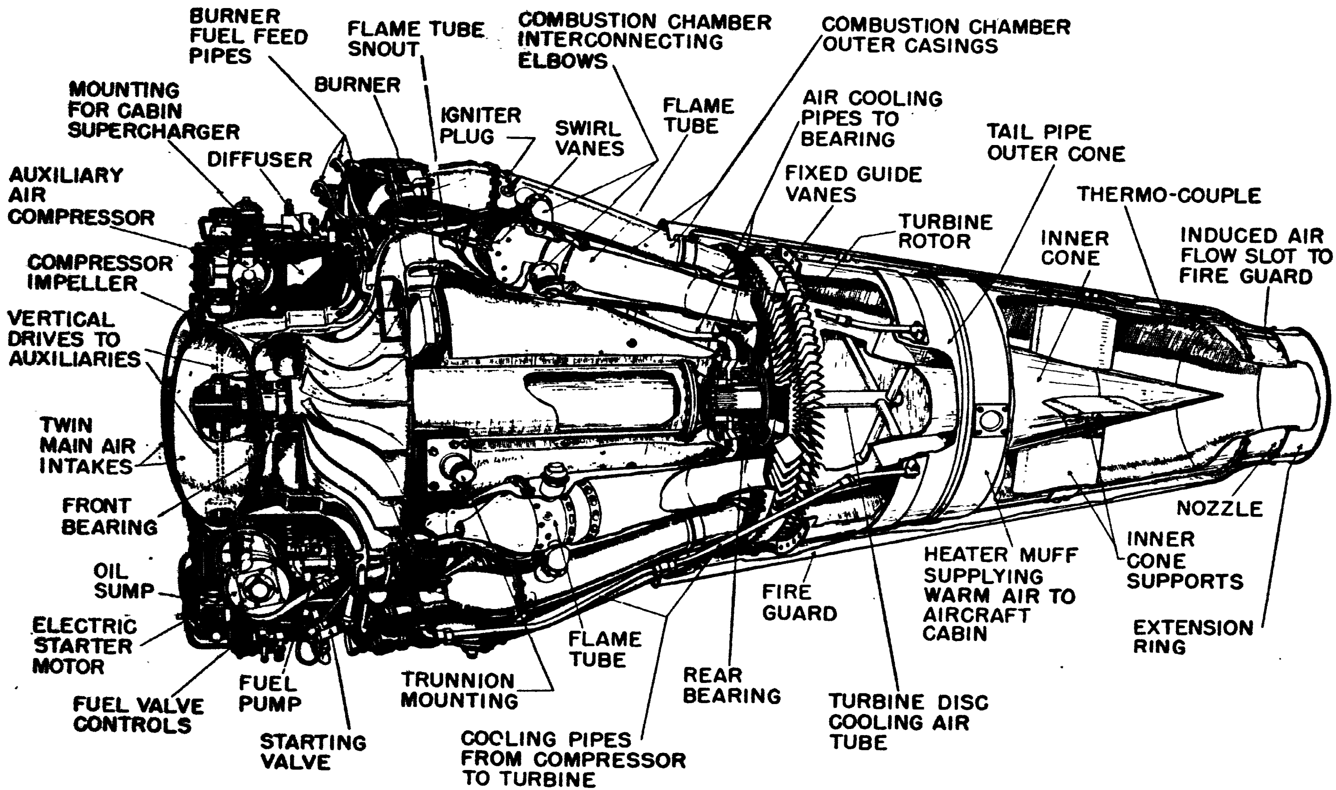
assembly, as compared with three required with a double-sided impeller. Second, that the axial load component on the turbine blades can be almost balanced by the forward-acting thrust loading on the impeller. The degree of unbalance is governed, within limits, by the positioning of the 14 labyrinth sealing rings on the back of the impeller, and today, the component of the pressure acting forward on the impeller rear face is equal to 20 psi. A final consideration in selecting the single-sided impeller was that the resulting complete power unit suited the layout of the Vampire, for which it was intended.

Before production of the Goblin prototype could begin, there remained one more problem to be settled: the detail design and selection of materials for the turbine disk and blades. Theoretical information was available on blade form, but little was known about the operating temperatures likely to be experienced. Chief properties required of the selected materials were a high resistance to creep, corrosion, and scaling. On the first few engines, turbine disks were made of austenitic steel, but later, with the aid of heat-sensitive paint, it was found that ferritic steels, with their greater strength around the working temperature of 750° C. and with a lower coefficient of expansion, would prove satisfactory. Operating temperatures



Courtesy of Flight Magazine

Fig. 1. Series II Goblin.

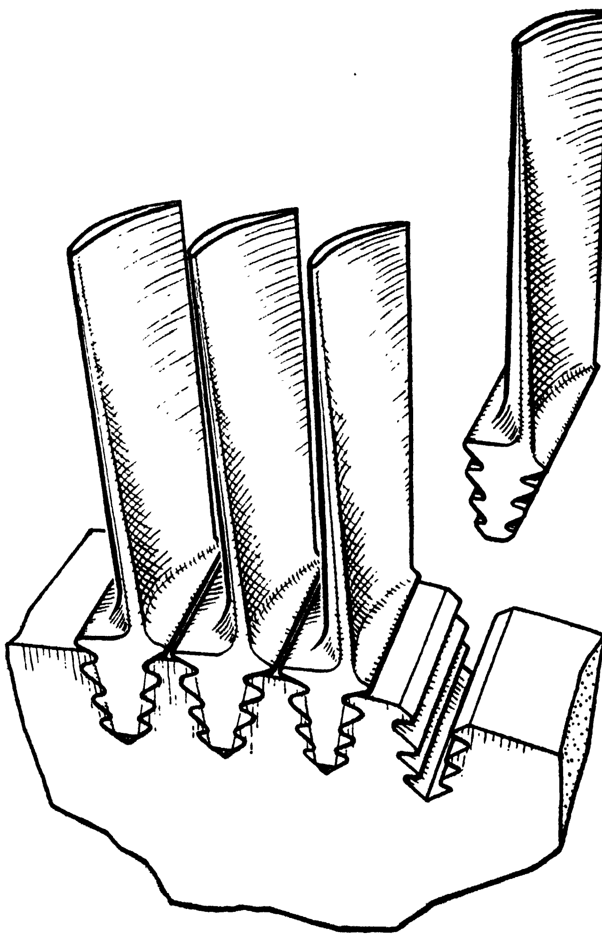


Courtesy of Flight Magazine

Fig. 2. Series II Goblin—Power Unit of the Vampire Fighter.

and materials which now apply to the Goblin are given in Tables 1 and 2.

By June 1942, the Goblin was tested up to its designed

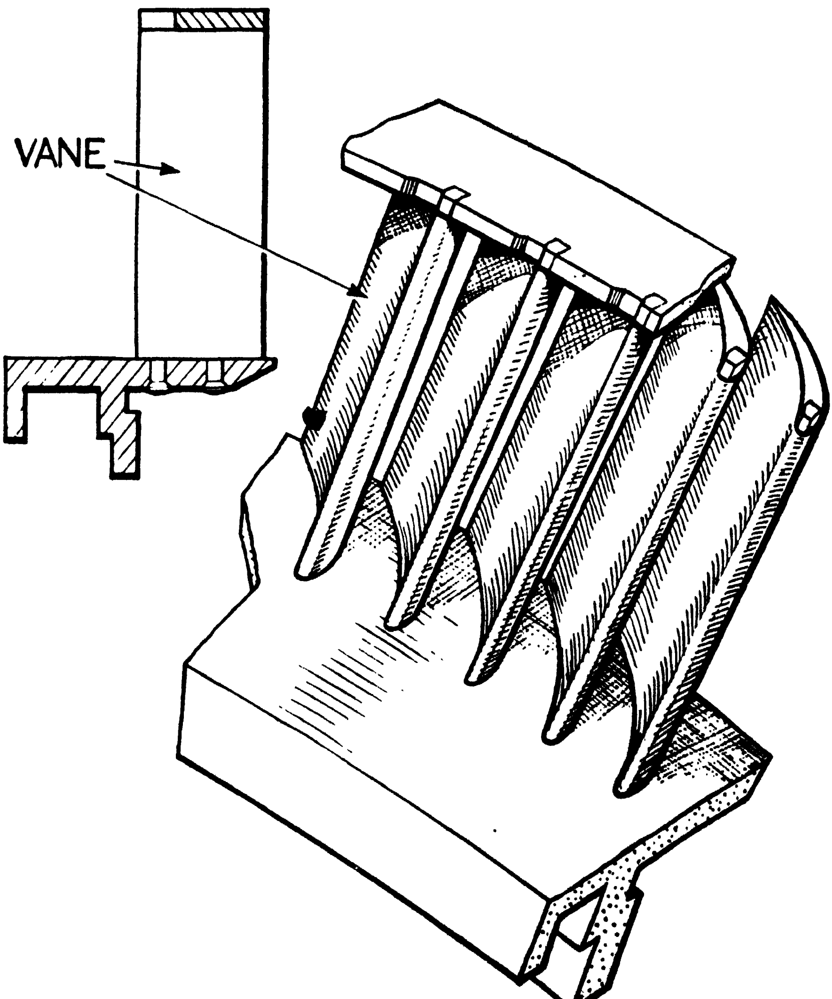


Courtesy of Flight *Magazine*

Fig. 3. ***Fir-tree*** **Turbine-blade Attachments.** Taken by permission from *Gas Turbines and Jet Propulsion for Aircraft,* published by Aircraft Books, Inc., N. Y. C.

rating of 3000 lb thrust at 10,500 rpm. The tail-pipe temperature was around 620° C and the specific fuel consumption 1.233 lb per lb thrust per hr.

A single-stage, single-entry centrifugal compressor provides an air flow of 60 lb per sec at maximum operating speed. The power required to drive the compressor is a little under 6000



Courtesy of Flight *Magazine*

Fig. 4. The Fixed Nozzle Blades Are Secured by Two Pins and One Peg.

hp. At 10,500 rpm the gyroscopic couples are about half those for a comparable piston-engine propeller installation. The impeller is a one-piece, heat-treated light-alloy anodized and pol-

ished forging. It is 31 in. in diam, has 17 vanes, and reaches a maximum tip speed of 1430 fps.

The gas turbine which provides driving power for the compressor and auxiliary drives is of the single-stage axial-flow type. It has 77 stator and 83 rotor blades, these numbers having no common denominator to avoid resonance. Tip diameter is about 27 in. and the blades, tapering slightly from root to tip, vary between 1.3 to 1.1 in. in width. Under operating conditions, the centrifugal stress at the blades is 20,000 psi and the maximum gas bending stress is 28,000 psi. The constructional material, Nimonic 80, is an alloy with very high nickel content. The creep limit in the ferritic steel turbine disk is 0.1 per cent, but this has not in fact been approached on components checked after the 300-hr running period.

The moving blades are attached to the turbine disk by the *fir-tree* method. The blades are held in position by peening at the roots on each side. On the upstream side, the peening is a little heavier to resist rearward thrust (Fig. 3). The fixed nozzle blades are shown in Figure 4.

The large-diameter center shaft is machined from a steel forging, which is carried in two ball bearings. Thrust loading is taken by the front bearing which is located on a stub shaft between the air-intake ducts in front of the impeller. This stub shaft is shrunk into, and bolted onto, the impeller. All the bolts for the attachment of both stub and main shafts pass through tubular dowels which take all shear loading, the studs being in tension. Drives for the auxiliaries are taken from the stub shaft through bevel gears.

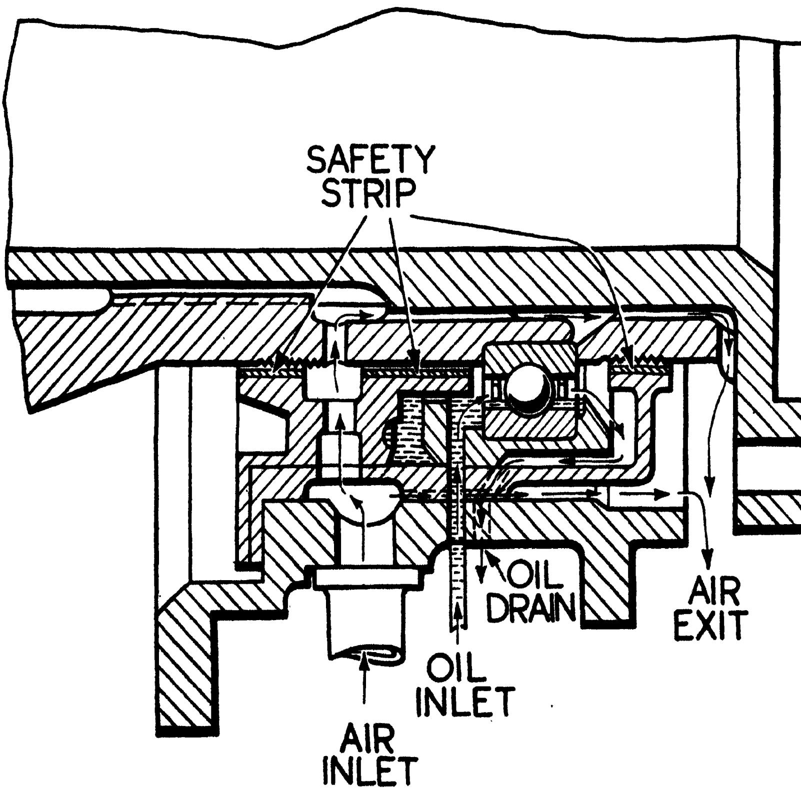
The rear bearing is fitted in a special sliding housing and is designed to take only journal loading (Fig. 5). It is air cooled, carries a heat-insulating plate, and runs at a temperature of about 130° C. The cooling air circuit and the lubrication are described below.

Turbine and impeller are first balanced separately, and then as a unit. The limits permitted are 5 to 7.75 g-in., and subsequent measurements of vibration for the whole power unit have shown that the vertical oscillation is less than ±0.001 in.

Improved Combustion Chambers

Sixteen combustion chambers are fitted to the Goblin series units.

The bulk of the intake air is admitted to the flame tube in stages as it passes round between tube and outer casing. Its



Courtesy of Flight *Magazine*

Fig. 5. Details of the Sliding Rear Bearing.

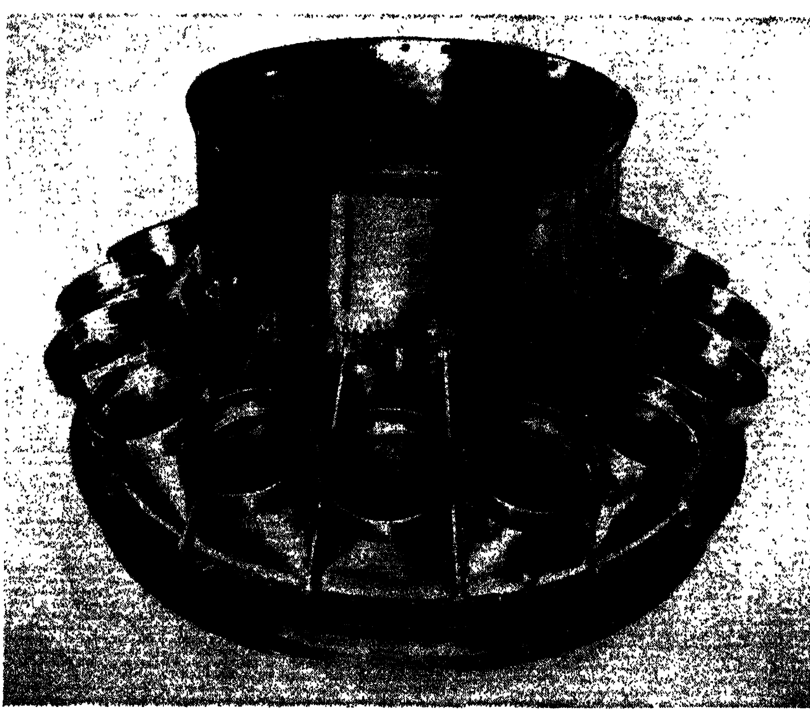
purpose is dilution and cooling of combustion gases which. from an initial 2000° C near the burner, enter the turbine blades at not more than 790° C. Combustion is completed within one third of the length of the tapered section of the flame tube.

With regard to the fitting of combustion chambers, the top end is in each case bolted to a small light-alloy expansion

chamber, and this is in turn bolted to the diffuser casing. At the lower end, to allow for expansion, there is a piston-ring joint where the outer casing slides into a seating in the nozzle junction box (Figs. 6, 7, and 8). The flame tube inside the combustion chamber is a loose sliding fit at its tapered downstream end and it is positioned by six small depressions. At the top, it is located by three equally spaced radial pins registering with sockets on the flame tube. Interconnecting passages, made up of inner and outer tubes, maintain a balance of pressure between individual combustion chambers.

The Main Frame

The main frame of the Goblin comprises the compressor front casing with twin air intakes; the two-piece diffuser



Courtesy of **Flight** *Magazine*

Fig. 6. The Nozzle Junction Box with Skirt, Diagonal Stays, and Combustion-chamber Seatings.

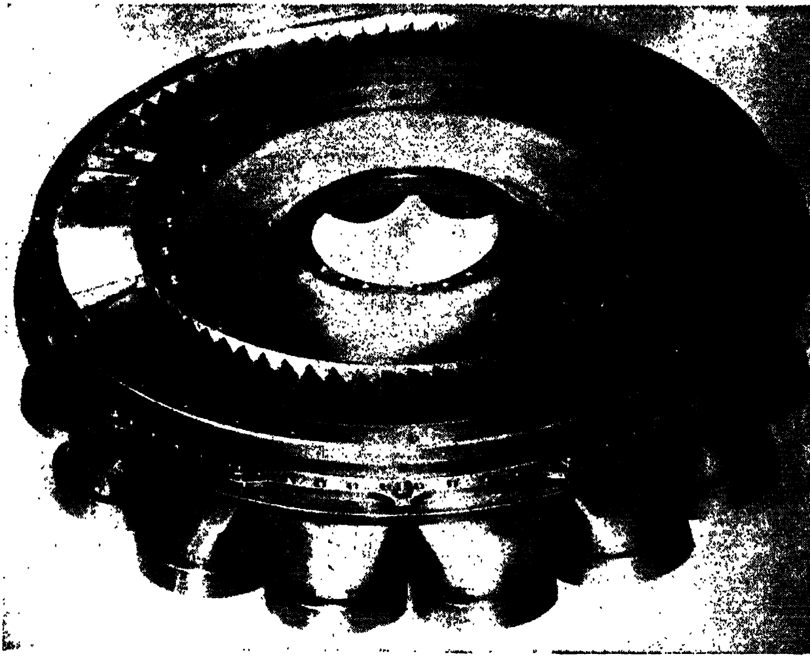


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Fig. 7. The Fixed Nozzle-ring Assembly Showing the Diaphragm for Radial Location.

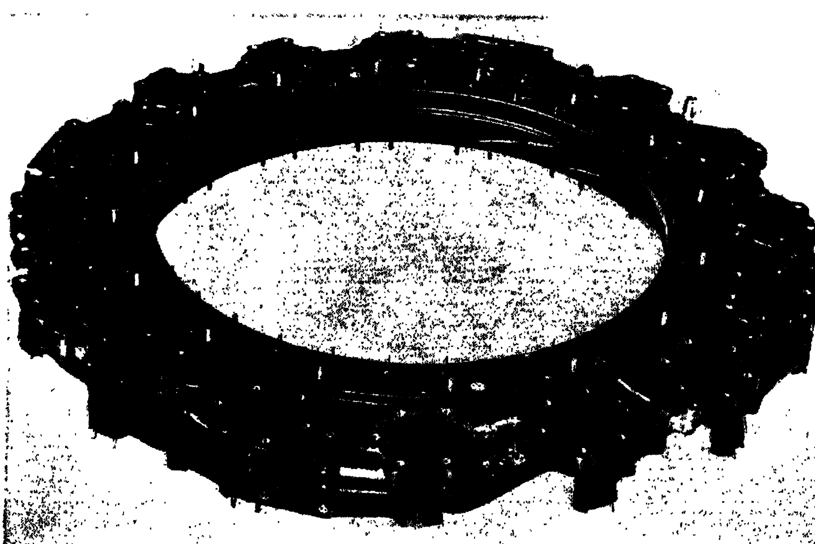

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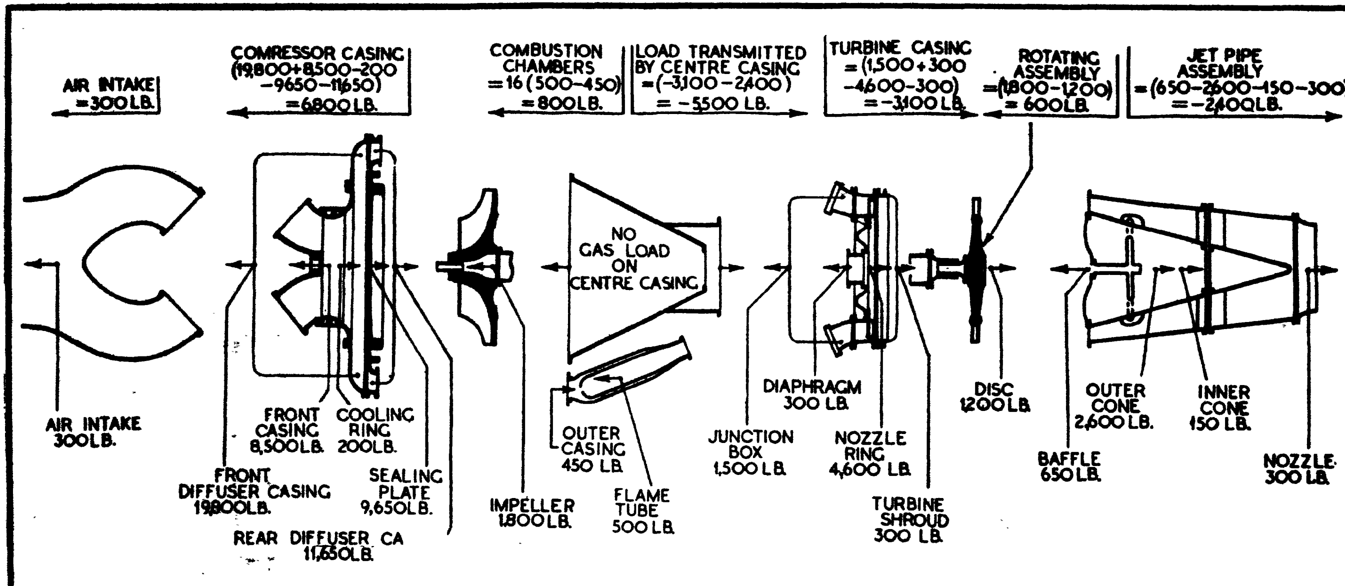
Fig. 8. The Diffuser Casing Is a Magnesium Casting Made in Two Halves.



Courtesy of Flight *Magazine*

Fig. 9. Component Axial Loads Under Static Conditions at Sea Level.

casing; the conical center casing with skirt; and the nozzle junction-box assembly with diaphragm plate and turbine bearing housing. Constructional materials for all main components (Figs. 2 and 9) are given in Table 2. The components are bolted together to form a rigid backbone for the power unit, and the nozzle-box assembly is further supported by diagonal stays to the center casing which isolate thrust, torque, and bending loads from the combustion chambers.

The diffuser casing is made in two halves and carries the passages and pipes for supplying cooling air to the rear bearing and turbine. There are two separate cooling air systems, but the air for each is taken from the main supply from the compressor. The turbine rear face is cooled by an air stream between the disk and rear baffle plate. Air is taken from four points in the rear diffuser casing near the tips and is led in separate pipes through the streamlined inner-cone supports to the back of the turbine. For the turbine bearing and front face, air is supplied through holes in front of the diffuser casing, and is led through pipes to a cooling ring in the main air-intake casing. From this point, the cool air is led to the turbine sliding bearing and finally flows up the front face of the turbine disk, which is thus cooled on both sides.

The cooling ring mentioned is sweated into the intake casting at an early stage and is machined down to give a perfectly smooth duct interior. Its presence can only be detected by an enlarged band on the outside of the casing and by the pipes leading to it.

Bolted to the rear bearing housing is a deep-drawn stainless-steel diaphragm plate which gives radial support to the static part of the turbine assembly. The main components of this assembly are the nozzle junction box, the fixed nozzle ring, and the stationary turbine shroud. The method of attaching the stator blades by two pins to their inner ring and by a single peg to their outer ring is illustrated in Figure 4; the outer ring is split into segments to allow for expansion.

After leaving the turbine, the combustion gases pass through the tapering jet pipe to the propelling nozzle. Supported by four 90-deg streamlined stays in the jet pipe is the inner cone,

or bullet, and around the jet pipe, in two main sections, is the lagging. Between lagging and jet pipe is a narrow space through which flows air for heat insulation; the air is also available for aircraft heating purposes.

Normally, fuel is supplied by an engine-driven pressure pump of the rotary type, having seven plungers reciprocating in a cylinder block that rotates round a central stationary shaft. The pump is rated at 650 gal per hr at a pressure of 800 psi at 3500 rpm. This corresponds to an engine speed of 10,500 rpm. Before delivery, the fuel passes through a control box containing a metering orifice the area of which is controlled by a tapered needle operated from the pilot's throttle lever. The contour of the needle is arranged to provide a roughly linear relationship between engine thrust and throttle lever travel. A barostat automatic altitude control gives a constant engine speed, without manual adjustment, between sea level and 38,000 ft.

The fuel used is aviation kerosene, which has a specific gravity of 0.05 at 15° C; a freezing point of — 40° C; a flash point of 30° C; and a spontaneous ignition temperature of 450° C.

Oil is delivered by a gear-type pressure pump at 50 psi. The sump has a capacity of 12 pt, and is mounted on the front casing with the accessories. All accessory drives are carried in ball bearings and some 90 gal of oil per hour are delivered to the auxiliary drives for which the main driving gears are lubricated by seven high-pressure jets. Oil is delivered to the two main bearings at the rate of ½ pt per hour, through metering pumps, and the pilot receives visual warning if the oil pressure drops below or rises above the limits permitted.

The starting cycle is automatic, and the electric motor, which is geared to run at about three times engine speed, develops 9 to 12 hp. To obtain the necessary air flow, the impeller must be turned at 900 rpm, and for satisfactory acceleration to idling speed, 1500 rpm is required. After closing the safety switch in the cockpit to energize the starting and fuel booster-pump circuits, the starting sequence is as follows: With throttle lever at the slow running stop, and high-pressure fuel-distri-

bution valve set at *open,* the starting button is pressed. The motor is initially energized through a high resistance which keeps the speed down, and permits shock-free engagement of the starter dogs. After some six seconds, a time switch cuts out the resistance, and the motor accelerates until 600 rpm is indicated for the engine. During the next 17 sec, the engine pump charges the fuel accumulator to the starting pressure of 40 psi, and the spring-loaded diaphragm of the automatic starting valve is then forced off its seat and held in the open position by a mechanical catch. Fuel is now permitted to flow to the burner ring, and as soon as a pressure of five pounds per square inch has built up, the automatic pressure switch is pressed, and full current is supplied to the starting motor. The starter motor now increases the indicated rpm to about 1500 during the next 13 sec, and the two igniter plugs, fitted with their electrodes across the fuel spray, light up the fuel in the combustion chambers, the flame traveling quickly round to the other combustion chambers through the connecting pipes.

The starting cycle is completed when the rpm have risen to 3000, and the pilot takes over from this point. About three minutes of slow running at 5000 rpm is recommended before opening up to take-off conditions. All throttle movement should be made slowly, and the time taken to increase the engine speed from idling to maximum rpm should be not less than 10 sec.

The unit should be allowed to idle and cool off for about half a minute before shutting down. The high-pressure fuel-distribution valve should then be closed, and this action causes the drainage of the fuel system and the release of the safety catch holding open the starting-valve diaphragm.

In emergency, it is possible to stop the engine quickly by simply cutting off the fuel supply, but this results in the deluging of very hot components with quantities of cold air, and the drop in temperature may cause damage.

If a Goblin has for some reason stopped in the air, restarting may be attempted at an engine speed of about 1200 rpm, but this is an emergency operation.

The following engine instruments are provided in the cockpit in addition to the oil-pressure, warning light already

TABLE 1

D. H. GOBLIN II—PERFORMANCE, DESIGN DATA, AND DIMENSIONS
10,200 RPM STATIC SEA-LEVEL CONDITIONS

Fuel flow	5,720 lb per hr
Maximum fuel pump delivery pressure	500 psi
Blower pressure ratio (over-all)	3.3 to 1
Blower pressure ratio (intake-delivery)	3.75 to 1
Blower temperature rise	150° C
Blower horsepower	5,720
Air mass flow	60 lb per sec
Air fuel ratio	58 to 1
Turbine inlet temperature	790° C
Jet velocity	1610 ft per sec
Maximum jet pipe temperature	685° C
Cruising jet pipe temperature	550° C
Climbing jet pipe temperature	630° C
Maximum rear bearing temperature	130° C
Oil consumption maximum, all conditions	1.5 pt per hr
Normal oil pressure (cruising)	40 to 45 psi
Oil tank capacity	12 pt
PERFORMANCE	
Maximum static thrust	3,000 lb at 10,200 rpm
Cruising static thrust	1,850 lb at 8,700 rpm
Idling static thrust	150 lb at 3,000 rpm
Maximum burner pressure at take-off rpm	650 psi
Minimum burner pressure at idling rpm	15 psi
Specific fuel consumption, take-off 1.23 lb/lb/hr	10,200 rpm
Specific fuel consumption, climbing 1.23 lb/lb/hr	9,700 rpm
Specific fuel consumption, cruising 1.30 lb/lb/hr	8,700 rpm
DIMENSIONS	
Maximum diameter	50 in
Length (engine air intake to propelling nozzle)	107 in
Propelling nozzle diam (internal)	16 in
Propelling nozzle length	6.25 in
Exhaust cone length (standard)	43 in
Impeller tip diam	31 in
WEIGHTS	
Rotating assembly	337 lb
Combustion chambers (16)	289 lb
Junction box	111.5 lb
Center casing	66.8 lb
Support cylinders	31.0 lb
Tail pipe	126.0 lb
Diaphragm and insulating plates	19.0 lb
Turbine bearing housing	19.5 lb
Turbine shroud	22.5 lb
Diffuser rear cover	103.0 lb
Diffuser casing	277.5 lb
Sundries	99.5 lb
Engine accessories	68 lb
Total weight	1,570.3 lb

TABLE 2

CONSTRUCTIONAL MATERIALS

Compressor impeller	R.R. 59 (L.42), drop stamping from continuous cast ingot (silicon 0.25 per cent max)
Turbine rotating blades	Nimonic 80, individual drop stampings
Turbine disk	Hecla 153, or Jessops H.3.A, drop stamping
Nozzle blades	Jessops G.18B, or Nimonic 75, drop stamping
Nozzle inner, outer, and shroud rings	H.R. Crown Max centrifugally spun castings, annealed
Junction pipe	Inconel from sheet
Nozzle assembly bolts	D.T.D.176A austenitic steel (with S.80 nuts)
Combustion-chamber front outer casing	Aluminum alloy D.T.D.272, annealed, die casting
Combustion-chamber rear outer casing	Mild steel deep drawing, protected by nickel plating
Combustion-chamber flame tube	Inconel from sheet
Center casing	Stainless steel (F.D.P.) from sheet
Jet-pipe assembly	F.D.P. from sheet
Diffuser and front casing	Magnesium alloy (D.T.D.281), castings annealed

mentioned: jet-pipe temperature indicator (zero to 800° C); turbine-bearing temperature indicator (zero to 200° C); a fuel pressure gauge (zero to 800 psi); rev counter (zero to 20,000 rpm).

Goblins are at present installed only in the Vampire single-seater fighter which is in production for the R.A.F., but trial installations in other aircraft, in England and America, have met with equal success, and have shown it to be a reliable and excellent turbine jet unit. In the Sea Vampire, the Goblin became the first pure jet unit to make a deck landing and take-off.

Chapter XIX

Armstrong Siddeley Turbines[1]

Prior to its entry into the gas turbine field, Armstrong Siddeley Motors Ltd. had many years of experience in the manufacture of automobile and aircraft engines of the more conventional piston type. However, this company had done experimental work on axial-flow compressors and blade forms. Early in 1942, the company for the first time seriously considered the production of gas turbine engines, and the design of the A.S.X. jet unit was subsequently prepared (Fig. 1). Later the A.S.X. was redesigned in order to develop a prop-jet unit.

The conversion from one to the other model chiefly entailed provision of a suitable reduction gear and the redesigning of the turbine to provide the necessary shaft horsepower. In other respects, the engine remained substantially unaltered, and under the name *A.S.P.*, or *Python,* first ran in March 1945 (Fig. 2).

The layout adopted, having the compressor intake well back from the propeller shaft, permitted good accessibility and an efficient intake.

The main static and moving components, in order of their positions from front to rear of the unit, are as follows:

Principal static *frame* units:

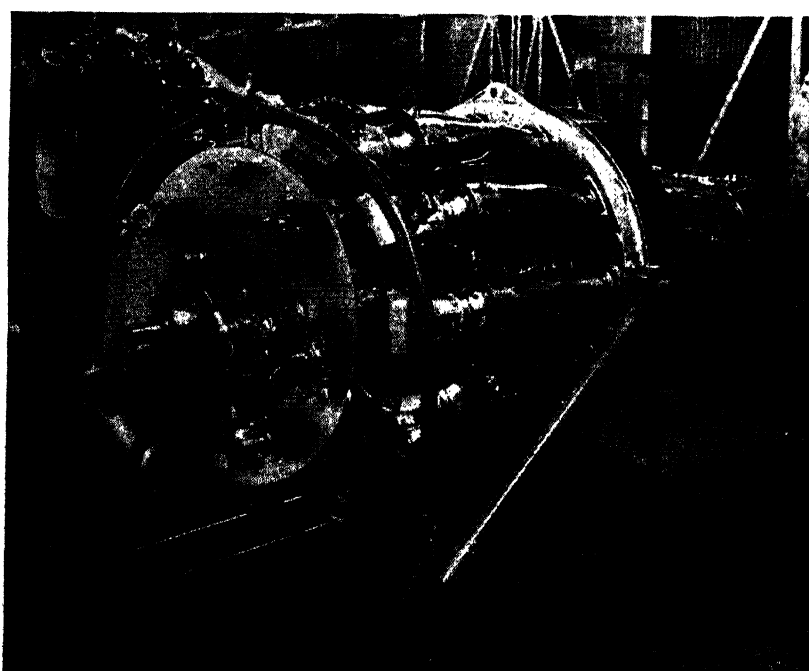
- front cover and accessory casing
- diffuser-casing support
- diffuser-casing cover
- diffuser casing
- main compressor or stator casing (split fore and aft)
- air-intake body
- air-intake throats
- back support plates with turbine stator brackets

[1] *Flight,* April 4, 1946.

manifold
first-turbine stator blade ring
first-turbine stator ring
second-turbine stator blade ring
second-turbine stator ring
exhaust cone
tail pipe
propelling nozzle

Principal moving components (excluding propeller driving and reduction gears):

compressor front main shaft
compressor high- and low-pressure drums
rear main shaft
stub shaft
turbine disk



Courtesy of Flight *Magazine*

Fig. 1. Armstrong Siddeley A.S.X. Engine with 14-stage Axial-flow Compressor and Two-stage Turbine.

In addition to the units named, there are the combustion chambers and their extension, or manifold pipes, which pass between the intake throats and the manifold (nozzle ring) of the turbine.

The main rotating drum of the A.S.X. compressor consists of two forged aluminum-alloy sections (R.R.56) bolted together on the inside, the division being between the high- and low-pressure stages. Of the 14 stages, five are low-pressure and nine, high-pressure. The outer casing, made in two parts, is a light-alloy casting, split horizontally and carrying the fixed blades.

Gas-flow diagrams for the A.S.X. and Python are similar. During its passage from intake to exhaust, the air flow is reversed twice in direction and passes in turn through the following components:

1. Air enters cowling around the unit as shown in Figure 3. Cowling is removed from Figure 2.

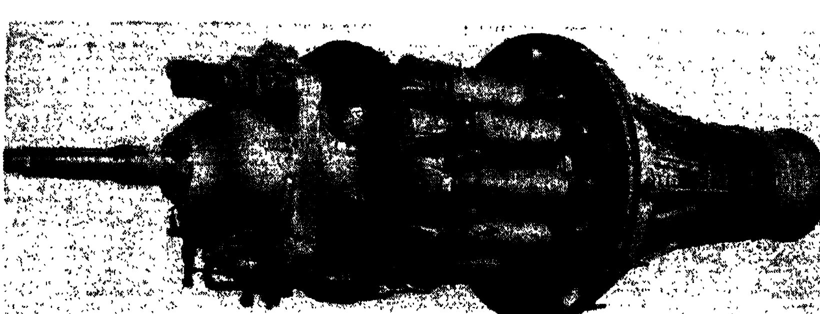
2. It then enters rear end of compressor through ducts.

3. It leaves the front of compressor, taking a sharp turn into the combustion chambers.

4. Passing through the combustion chambers it goes to the rear of the unit and passes directly into the two stage turbine.

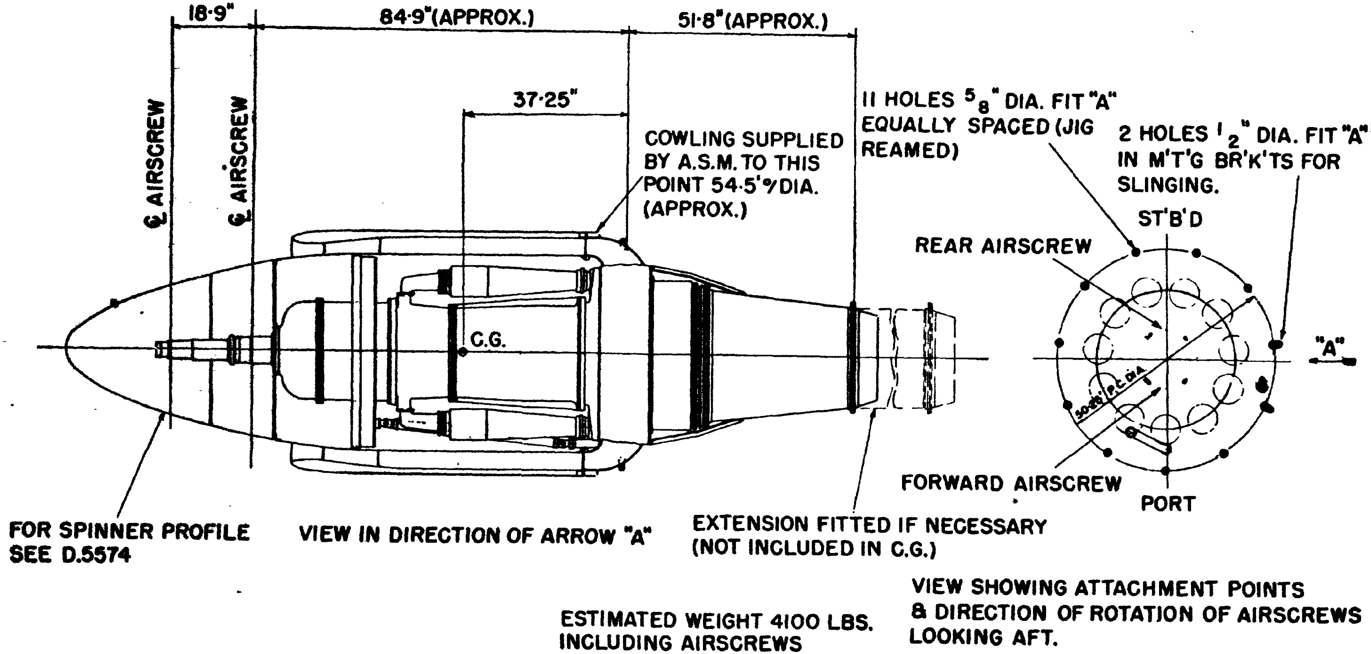
5. What energy is left is expanded out the jet.

On each elbow to the combustion chambers, there is a blow-off valve, the purpose of which is to aid in starting by releas-



Courtesy of Armstrong Siddeley

Fig. 2. Armstrong Siddeley Prop-jet Python.



Courtesy of Armstrong Siddeley.

Fig. 3. Outline of Armstrong Siddeley Python S.P.1-1 Unit.

Courtesy of Flight *Magazine*

Fig. 4. The Universal Test Bed: Lancaster with A.S.X. Engine Installed in the Bomb Bay with Doors Open.

Courtesy of Armstrong Siddeley

Fig. 5. Armstrong Siddeley's Small Prop-jet, the Mamba.

TABLE 1

A.S.X. AND PYTHON DATA

A.S.X.

DIMENSIONS

Max diam	42 in
Over-all length (to propelling nozzle)	14 ft

WEIGHTS

Net dry weight	1900 lb

PERFORMANCE

Max speed of rotor	8000 rpm
Static sea-level thrust (take-off and combat)	2600 lb
Fuel consumption at take-off rpm	1.03 lb per hr per lb thrust
Fuel consumption at cruising rpm	1.0 lb per hr per lb thrust

A.S.P.I. (PYTHON)

DIMENSIONS

	S.P. 1-1	S.P. 1-2
Max diam over cowling	54.5 in.	48 in.
Over-all length over cowling	11 ft	4 in.

	S.P. 1-1	S.P. 1-2
Length mounting flange to C/L rear propeller	7 ft 1 in.	7 ft 9.5 in.
Annular air-intake duct area	2.5 sq ft	...
Propelling nozzle area	2.6 sq ft	...

WEIGHTS

	S.P. 1-1	S.P. 1-2
Estimated net dry weight (including starter and accessory, box drive shaft)	3010 lb	2980 lb
Estimated installed weight (with propeller)	4100 lb	3950 lb
Fuel oil consumption	...	0.5 gal per hr
Lubricating oil circulation	...	1200 gal per hr

PERFORMANCE

Rating	Engine Speed Rpm	Aircraft Speed Mph	Propeller Shaft Hp	Net Jet Thrust Lb	Fuel Gal Per Hr
Max take-off, max combat	8000	0	3670	1150	359
		200	3950	760	372
		300	4290	590	385
		400	4860	420	401
		500	5520	280	425
Max climb	7800	0	3150	1060	323
		200	3450	660	334
		300	3800	490	346
		400	4200	320	361
		500	4950	170	384
Max cont. cruising	7600	0	2720	950	290
		200	2960	570	300
		300	3260	390	308
		400	3680	220	325
		500	4250	60	341

ing to atmosphere a proportion of the air passing through the compressor. The valves also give some aid to acceleration after starting by preventing the compressor blades from stalling. They are controlled by a lever in the cockpit. On the Python, these separate blowoff valves have been abandoned in favor of a single common valve at the front of the compressor casing.

The constructional material of the 11 combustion chambers is thin-gauge stainless steel and the flame tubes inside are of Nimonic 75. Of Armstrong Siddeley design, they differ considerably from others based on designs developed by the Lucas Company, and employ the principle of vaporizing of fuel rather than its atomization by means of high-pressure spray jets. The soundness of the principle has been proved thus far by generally trouble-free running up to the highest altitudes yet flown on the A.S.X. unit.

The Python has a single-lever cockpit control for the whole of the power range from slow running to full power, and each position of the pilot's lever both selects the appropriate propeller pitch and assures that the correct quantity of fuel is fed to the engine. Corrections for altitude, forward speed, and air temperature are made automatically.

When starting a large gas turbine engine starting power is required for a relatively long period. For this reason a specially designed gas starter motor is employed on the Python. On the A.S.X., an electric starter motor geared to the front extension shaft is generally used.

Most of the accessories face forward on the accessory casing around the reduction gear.

The Python reduction gear unit is of necessity rather bulky in that an over-all ratio of 0.123 to 1 is required in addition to the gearing for contrarotation of the propellers.

The essential engine auxiliaries are:

gas starter motor
combined oil-pressure and scavenge pumps
combined auxiliary oil-pressure metering and
  auxiliary scavenge pumps
constant-speed unit

electrical tachometer
fuel pump and overspeed governor
fuel-control barostat
fuel filter
high-pressure shut-off valve
igniter-spray jet control valve
blow-off-valve control system
automatic exhaust-temperature controls
ignition coils (2)

In addition, a shaft is provided for driving a remote accessory box at 0.326 times engine speed. At cruising rpm, the capacity is 60 hp.

Engine instruments required are:

rpm indicator
jet-pipe thermometer
oil thermometer
two oil-pressure gauges
fuel-distributor pressure gauge
torque-meter pressure gauge and fire indicator

Chapter XX

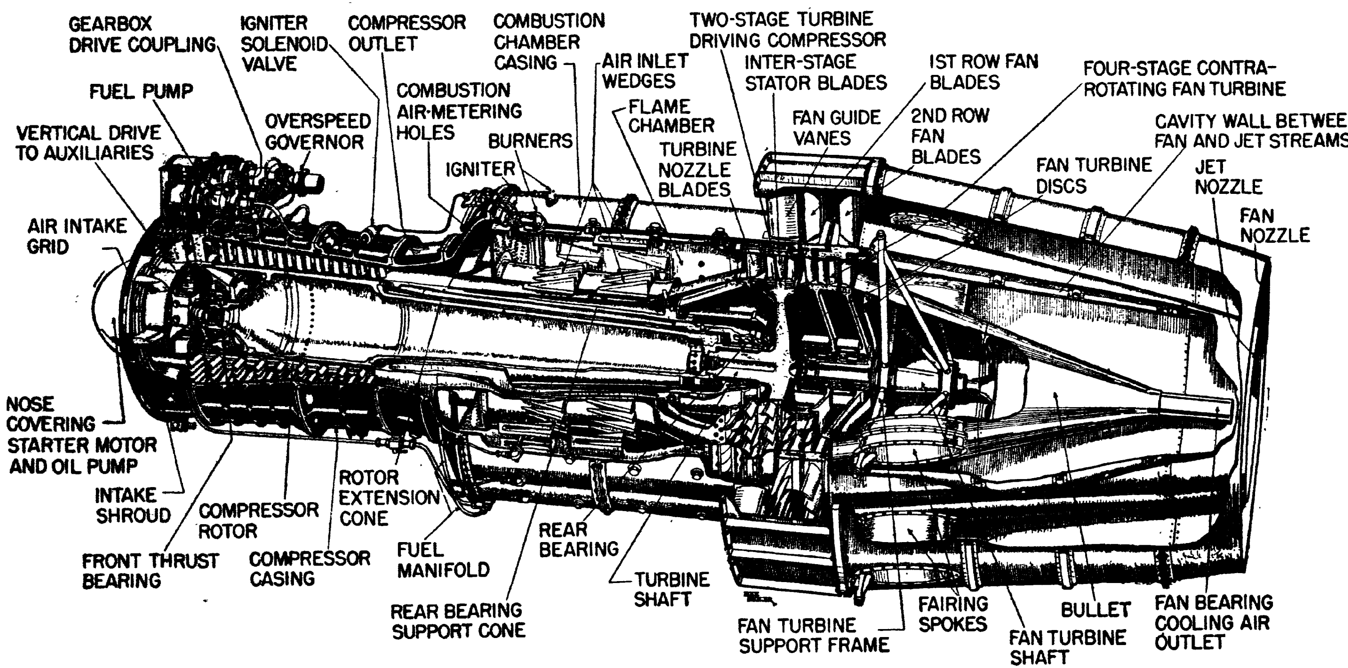
Metro-Vick Gas Turbine[1]

The unit which forms the subject of this chapter is a straightforward axial-flow compressor and gas turbine with which has been incorporated a ducted-fan thrust augmenter, known as the F/3 in Fig. 1. By this means, the thrust delivered by the unit has been increased by 67 per cent for an unchanged fuel consumption and for an increase in weight of only 33.3 per cent. In specific terms, the plain jet part of the unit produces a static thrust of 2400 lb for a weight of 1650 lb, giving a thrust-weight ratio of 1.455 to 1; but the unit, when fitted with the augmenter, gives a static thrust at sea level of 4000 lb for a weight of 2200 lb—that is, a thrust-weight ratio of 1.818 to 1, which shows a percentage specific gain of 25. This achievement is paralleled, if not eclipsed, by the reduction in specific fuel consumption. What is important about the embodiment of the thrust augmenter is that the gain in thrust is realized without increase in fuel consumption. The specific consumption, therefore, comes down to 0.65 lb per hr per lb thrust, which is approximately 35 per cent lower than that hitherto regarded as best obtainable, namely, 1 lb per hr per lb thrust.

The best way of visualizing a ducted-fan thrust augmenter in general terms is to regard it as a half-way measure between pure jet propulsion and prop-jet propulsion.

The streams from the gas turbines and the fans are separated and do not mingle until both have left the jet pipe (Fig. 2). At the orifice there is a hot high-speed jet from the turbines and a cold lower-speed annular jet from the fans, the

[1] *Flight,* April 25, 1946.

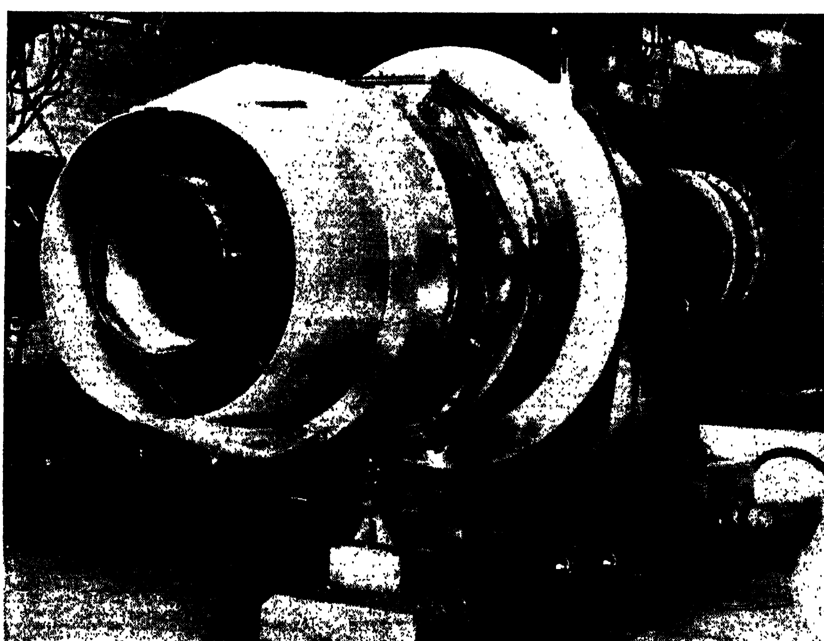


Courtesy of Flight *Magazine*

Fig. 1. Metro-Vick F/3 Gas Turbine.

latter stream cylindrically enclosing the jet stream. This arrangement has a most marked effect on noise suppression, and the unit is a very quiet aircraft propulsion engine. On this count alone, aside from the inherent lack of vibration and extremely good performance, its future for passenger aircraft should be promising.

Flow through the unit is accomplished by the air entering the intake to the compressor through a concentrically ringed guard. There are 70 fixed and 68 moving blades on each stator and rotor stage of the compressor respectively. All the fixed blades are similar in detail design, as are all the rotor blades —that is, all blades of one type are identical in twist, chord, camber, and so forth. However, the fact that the moving blades are cropped progressively in stage gives them, in effect, a coarser pitch in the higher than in the earlier stages. The compression ratio is 4 to 1 and, since the final ring of blades is a



Courtesy of Flight *Magazine*

Fig. 2. Metro-Vick F/3 Ducted-fan Thrust Augmenter, Bench Testing.

stator, the flow from compressor to combustion chamber is truly axial.

The 56 blades of the first ring are fixed and so arranged as to form a nozzle ring around the moving turbine blades. The latter are mounted in two rows of 80 each on the turbine disk and are separated by a ring of 84 stator blades interposed between them. This, then, comprises the main turbine, the purpose of which is to drive the compressor, and the maximum speed of which is 7600 rpm.

Rearward of the main turbine is a dual turbine composed of two contrarotating wheels, which operate in the gas stream exhausted from the main turbine. The dual turbine drives the fans, which are also arranged in two counterrotating banks. Since the two rings of turbine blades on each disk are nested together, with one row of each interposed between the two rows of its neighbor, the unit can justifiably be regarded as a four-stage component. Since the nested blades counterrotate, the need for fixed rings as stator blades is eliminated.

Structure

Structurally, the build-up of the main elements can well be seen in Figure 1, but we may with advantage briefly survey the salient features of the structural composition. The casing of the compressor is an RR50 light-alloy casting in four parts, top and bottom, and front and rear; the former bolted together at horizontal-meeting faces, and the latter bolted together circumferentially in the transverse plane of the sixth rotor stage. Dovetail-section axial slots are machined in the bore of the casing, and the stator blades are held in these, spaced by distance pieces and locked by peening. Bolted to the front face of the compressor casing is the intake shroud which, by means of *spokes* across the intake annulus, supports a coned diaphragm in which the front thrust bearing is housed. Again, since the compressor blades are designed to *screw* air into the unit, they have a forward reaction, and since the turbine blades *windmill* in the gas stream and thus have a rearward reaction, the opposed thrust components of the respective blades tend

to compensate each other, and so the thrust loading on the bearing is reduced.

The compressor drum is a single-piece forging in the outer surface of which are machined axial, parallel serrated slots for anchorage of the rotor blades, these also being spaced by distance pieces and locked by peening. A coned diaphragm, bolted internally to the drum at the third stage station, is spigoted to run in the front thrust bearing, and internal splines in the bore of the spigot transmit the drive through a splined quill shaft for powering the auxiliaries. The starter motor engages the forward end of the quill shaft through a plain dogged coupling.

At the rear end of the compressor drum is bolted a conic extension, this having bolted to its rear end an internally splined sleeve which runs in the rear journal bearing and houses the splined shaft of the overhung turbine wheel. The journal bearing is housed within the rear end of another cone extension member (this one being coaxial with, and enclosing, the internal extension cone) which, at its forward end, is bolted to the rear face of the compressor casing and is vented at this point to form the compressor outlet branch.

Each of the twin disks of the fan turbine is separately supported on a pair of roller and ball thrust bearings, both sets of bearings running on a common shaft, which is carried in a six-point, triangulated star bracing of tubular members which also serve to support the fan ducting and jet tail cone, or bullet.

Chapter XXI

Rolls-Royce Nene I[1]

To supplement its own research, Rolls-Royce took over turbine development work from the Rover Company in April 1943, and after modifying and improving the Rover B-23 design, the latter became the Rolls-Royce Welland (Fig. 1). This first member of the River Class was a reverse-flow design of 43 in. diam and 850-lb wt. The turbine unit flew in the Gloster E.28, and after further development, it was conservatively rated at 1450 lb thrust and became the power unit for the twin-engined Meteor prototype, the Gloster F9/40.

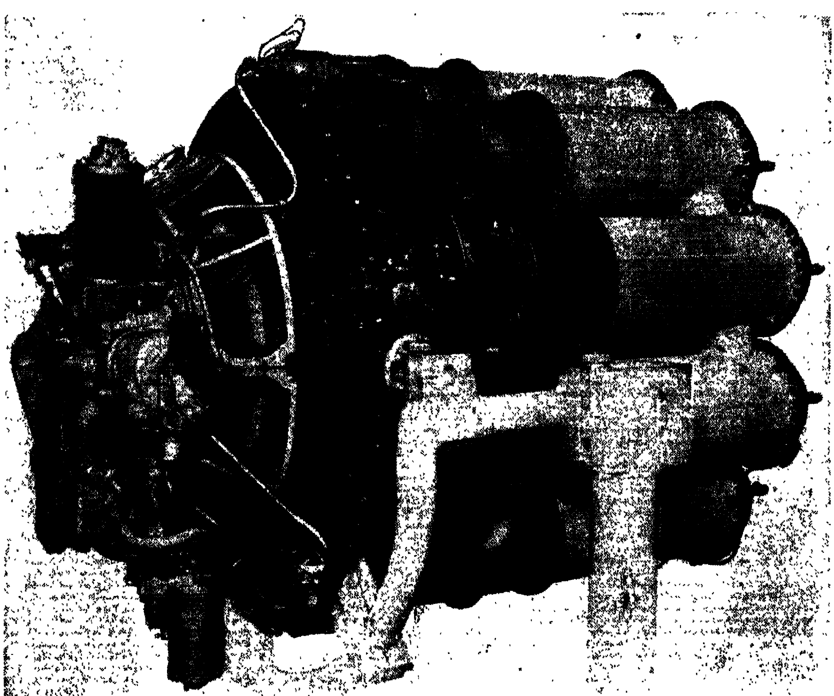
The Derwent, the second River Class engine, had by this time been further developed by Rolls-Royce, the main difference lying in the adoption of straight-through combustion as shown in Figure 2. Subsequent marks of the Derwent, the II and IV, each showed about 10 per cent thrust increase over the designed 2000 lb, and the Derwent V (Fig. 3), which later powered the record-breaking Gloster Meteors, passed the official 100-hr type test at a rating of 3000 lb thrust. As an indication of simplicity of design as compared with the piston engine, it is noteworthy that the complete Derwent V, which is different from the earlier Derwents, was produced straight from the drawing board, and it was the first engine off that passed the type test. Since that time, the Derwent V has undergone intensive development and now gives 4000 lb thrust.

The most recent and powerful addition to the River Class of turbo-jets, the Nene, is a turbine unit specifically for jet propulsion (Table 1 and Figs. 4 and 5). In all main design features, it resembles the smaller Derwent jet engine, but it

[1] *Flight*, April 18, 1946.

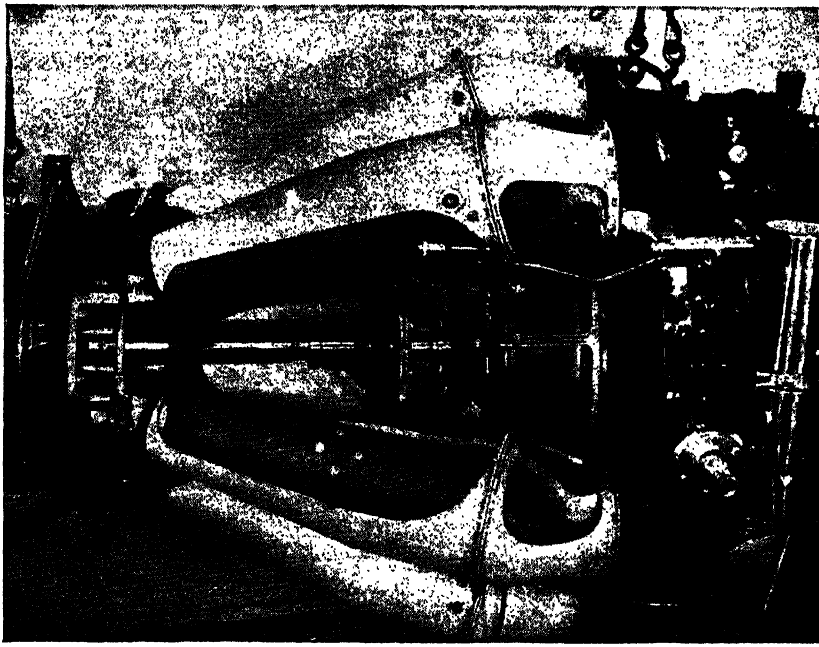
would be incorrect to regard it merely as a lineal descendant of increased dimensions. When the specification was first presented, the Derwent had been developed from the Mark I, producing 2000 lb thrust as compared to the Mark IV which had a thrust of 2400 lb. The practicability of scaling up the Derwent was investigated, but to obtain the stipulated 4000 lb thrust would have necessitated increasing the maximum diameter from 43 to nearly 60 in. Consequently, a complete redesign was undertaken, and it was found that all requirements could be met with a diameter of only 49½ in.

In the remarkably short period of 5½ months, the design was completed, all drawings prepared, the first unit built, and the proving run of one hour at 5000 lb thrust successfully ac-



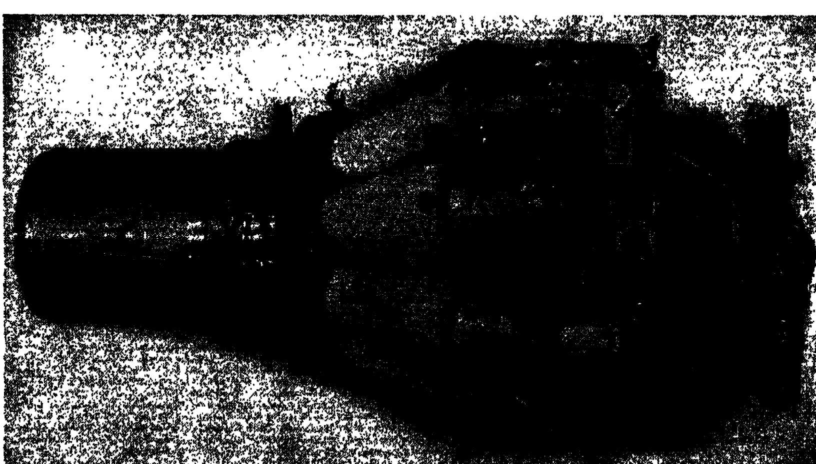
Courtesy of Rolls-Royce Ltd.

Fig. 1. Rolls-Royce Welland—the First Rolls Royce Development of the Whittle Engine. Pure jet unit, 1-stage centrifugal compressor, 10 reverse-flow combustion chambers, 1-stage axial flow turbine.



Courtesy of Rolls-Royce Ltd.

Fig. 2. Rolls-Royce Derwent I—1-Stage Centrifugal Compressor, 1-Stage Axial-flow Turbine. Ten combustion chambers differ from "Welland" in that they are straight-through type. Small fan visible on impeller shaft in front of thrust bearing cools bearings and turbine rotor.

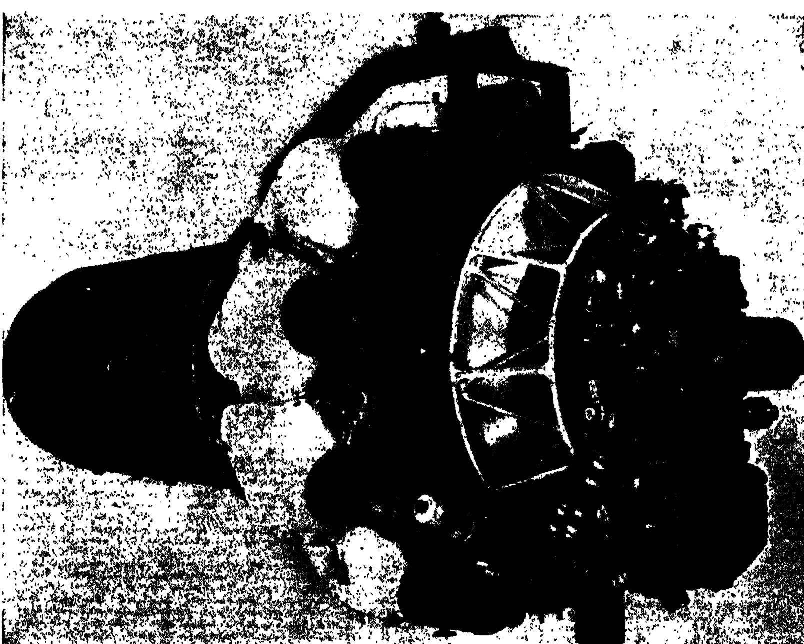


Courtesy of Rolls-Royce Ltd.

Fig. 3. Rolls-Royce Derwent V—Engine Installed in Record-breaking Gloster Meteor IV. One-stage centrifugal compressor, 9 straight-through combustion chambers, 1-stage axial-flow turbine.

complished. It is understandable that in view of this achievement, further development of the original Derwent design was regarded as relatively unfruitful, and, accordingly, the Derwent V was virtually a scaled-down version of the Nene I.

From the rear ends of the nine combustion chambers, the heated air and combustion products are directed by stationary, nozzle guide vanes on to the blades of the single-stage, axial-flow turbine rotor. Some of the energy in the gases is converted into mechanical power utilized to drive the compressor which is directly coupled to the turbine, and to maintain continuity of operation. From the turbine, the gases are discharged rearward through the exhaust cone and the jet pipe. The issuing



Courtesy of Rolls-Royce Ltd.

Fig. 4. Rolls-Royce Nene—Britain's Largest Jet Engine, Successor to the Derwent V. One-stage centrifugal compressor, 9 straight-through combustion chambers, 1-stage axial-flow turbine. Engine auxiliaries are mounted on the compressor casing directly in front of the compressor. Screens cover compressor inlets.

jet spreads at an included angle of 15-20 deg, and clearance must be provided.

The complete rotating assembly comprises the single-stage, double-sided, radial-flow compressor, cooling fan, and turbine rotor on two coupled shafts supported in three bearings. Compressor and turbine shafts are connected by a quick-detachable, spherically seated coupling which enables the turbine rotor and shaft to be withdrawn for inspection without dismantling the structure. The ball transfers axial thrust, but torque is transmitted by interengaging toothed elements.

Turbine

Jessop's B.18B steel is employed for the solid, forged turbine disk and, as on the Derwent, the blades are individually forged in Wiggins Nimonic 80 alloy. Blade roots are of the fir-tree type with tapered serrations fitting in complementary broachings in the rim of the disk and axially secured by peening. The shaft is not bolted directly to the disk, but to an integral flange spaced from the forward face in order to reduce the path by which heat can flow from the disk to the rear bearing. Serrations on shaft and disk flanges relieve the bolts of all shear stresses.

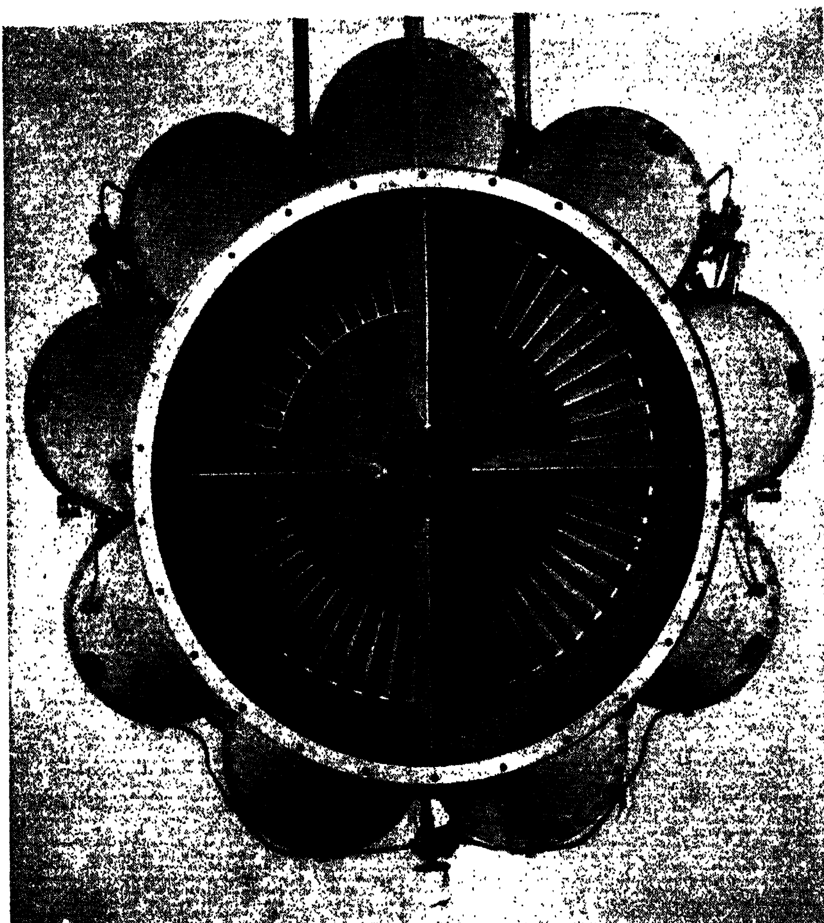
The nine combustion chambers, spaced evenly around the shaft, converge on the turbine nozzle box, which is a fabricated steel member consisting of nine circular-to-segmental sockets welded together and to an outer shroud ring and an inner ring, which supports the cooling air manifold. Later, a cast-iron nozzle box, similar to that used on Derwent V, will be provided. The 54 nozzle guide vanes are individually precision cast by the Austenol lost-wax process. When assembled, the guide vanes are gauged at five diameters to check the intervening apertures, and thus the total area of the inlet to the turbine.

The exhaust cone is bolted up to the turbine shroud ring and nozzle box with its inner cone supported by four transverse bolts. The base of the inner cone masks the rear face of the turbine disk. The exhaust cone is of fixed length, approxi-

mately 22 in., whereas the jet pipe extending from the exhaust cone to the propulsion nozzle can be varied to meet installation requirements.

Lubrication

Because there are only three main journal bearings and no sliding metal-to-metal surfaces, the lubrication system of a turbo-jet unit, as compared with that of a reciprocating en-



Courtesy of Rolls-Royce Ltd.

Fig. 5. Rolls-Royce Nene—Rear View. As shown, removal of the jet pipe permits visual inspection of turbine blades.

gine, is relatively simple. On the Nene, a wet-sump system is employed, constituting a change from earlier Rolls-Royce practice. The Derwent had a dry sump and separate oil tank.

Accessories

Conforming to current practice, aircraft accessory components are driven through a gear box mounted on the wheel case. A choice of three drives—horizontal, above, or below the main shaft and inclined from the head of the wheel case—is provided for a Rotol gear box weighing about 35½ lb. On this unit can be mounted and driven a hydraulic pump, electric generator, vacuum pump, and air compressor, as required for the installation. Gear box and accessories are all within the installation diameter of about 49 in. required for the Nene. Cabin heating or gun warming can be arranged by means of air heating jackets around the jet pipe.

A 24-v electric motor is used for starting, and all necessary operations are carried out by a timed automatic cycle.

The first aircraft to be powered by the Nene was a Lockheed X-P80 Shooting Star, and tests have also been conducted on a de Havilland Vampire. In both instances, an improvement in performance was obtained. New aircraft will be required to take full advantage of the thrust now available from the Nene. Several prototype aircraft, both single and twin-unit designs, are under construction.

The high thrust of 5000 lb and light weight of the Nene jet engine make it a potential competitor of the orthodox reciprocating engine for medium-range heavy bombers. Recently, Dr. S. G. Hooker gave estimated performance figures for a Lancaster powered with four Nene units. It was claimed that with an all-up weight of 60,000 lb, the cruising speed would be about 400 mph at all altitudes up to 35,000 ft. With existing fuel tanks, range would be approximately 1000 miles at 30,000 ft. Since the installed weight of the four Nene engines would be only 8000 lb as compared to 12,000 lb for four Merlins, tankage could be increased and the range extended. To have obtained a 25 per cent increase in power accompanied by a 33 per cent reduction in installed weight is an indication of

the importance of a change from reciprocating to rotating power units and a hint of the performance expected from specially designed aircraft.

Trent Turbine Prop-jet Unit

In addition to their development of the turbine-jet, Rolls-Royce has also produced an experimental turbine prop-jet unit known as the Trent, which was hangar-tested in March 1945. In September of the same year, it became the first unit of its

Courtesy of Flight *Magazine*

Fig. 6. Gloster Meteor Fitted Experimentally with Rolls-Royce Trent Engines. The Trent has been developed from the Derwent V and fitted with a reduction gear to drive a propeller shaft. Taken by permission from *Gas Turbines and Jet Propulsion for Aircraft*, published by Aircraft Books, Inc., N. Y. C.

kind to undergo flight trials. For this purpose, two were installed in a Gloster Meteor (Fig. 6).

Experimental work with this type of engine began in May 1944, when a compressor-turbine unit was equipped with a spur-type reduction gear and tested for shaft horsepower. The present Trent engine, which was produced solely for gaining experience with a unit combining jet and propeller propulsion, is still a Derwent with reduction gear and small-diameter five-bladed propeller added. Approximate weights during early stages of development are as follows:

Weight of turbine unit	1000 lb
Propeller	250 lb
Reduction gear	250 lb
Total	1500 lb

TABLE 1

NENE I DATA

Diam	49.5 in
Length, to turbine	63.9 in
Length, including exhaust cone	96.8 in
Weight, less jet pipe	1550 lb
Weight, jet pipe	9.5 lb per ft
Fuel—aviation kerosene + 1 per cent lubricating oil	0.806 S.G.
Fuel consumption, sea-level static, 4000 lb thrust	1.055 lb per lb thrust per hr
Fuel consumption, sea-level static, 5000 lb thrust	1.065 lb per lb thrust per hr
Air consumption, 5000 lb thrust	89 lb per sec
Compression ratio	4 to 1 static
Maximum speed	12,300 rpm
Maximum thrust	5000 lb
Compressor impeller, peripheral velocity	1530 ft per sec
Turbine rotor, mean blade speed	1070 ft per sec
Acceleration 2500–12,300 rpm	4.5 sec

Chapter XXII

Combustion and Fuel Research[1]

Pioneer Work of Lucas Laboratories

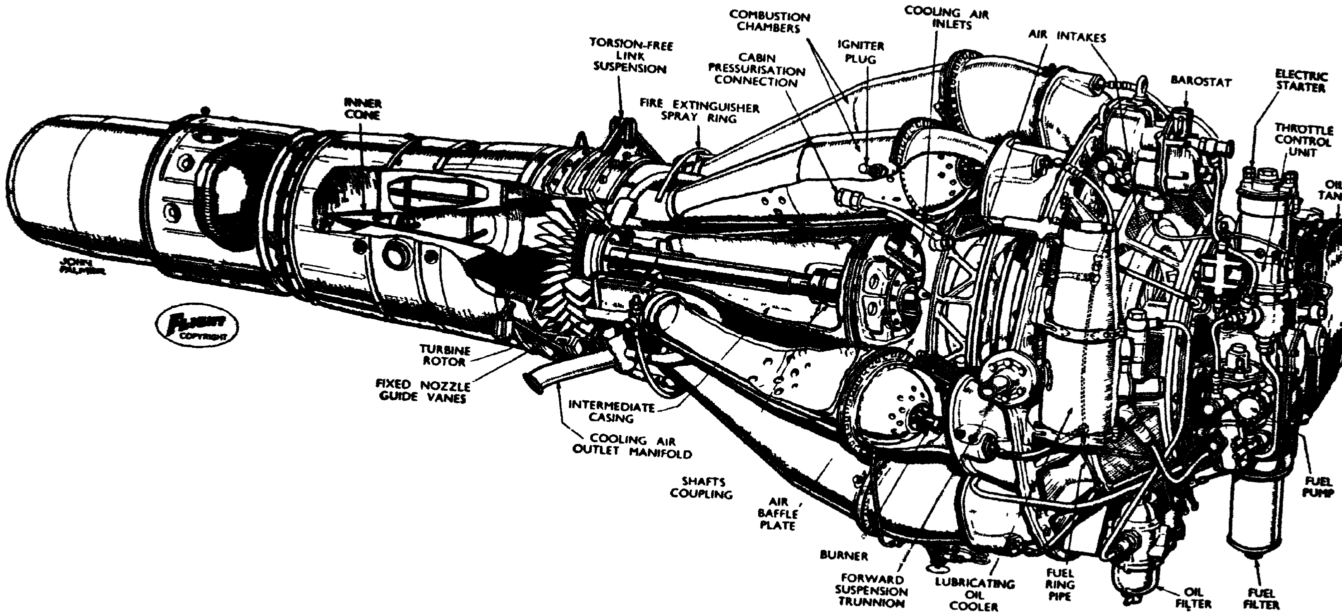
Combustion Research

In 1940, the newly formed Ministry of Aircraft Production approached Joseph Lucas, Ltd., to undertake a basic investigation in combustion and fuel systems, and the development and manufacture of the necessary equipment.

The immediate problem was to simulate conditions in the early Whittle engine, which required a supply of air at a pressure of 40 psi at the rate of 3.25 lb per sec for each combustion chamber. Even for a single-chamber test rig the air requirements were outside the scope of ordinary works supply, and a large compressor from the Ministry of Transport's equipment at the Dartford tunnel was obtained and installed at Birmingham. Driven by a 750-hp motor, this had an output of six pounds of air per second at 45 psi.

Initial investigations were to determine the characteristics and examine the behavior of the combustion chamber used on the Whittle W.2 and W.2B engines. Following this, a modified principle of primary air injection was developed, and after about three months' intensive work, was embodied in the W.2B engines constructed by the Rover Company. This basic design, known as the B.23 chamber, has since been applied to all subsequent models. When Rolls-Royce took over the engine development and manufacture early in 1943, it relied on the Welland engines. These engines were used in the units powering the Meteor aircraft which went into action against the

[1] *Flight,* Jan. 3 and 10, 1946.



Courtesy of Flight Magazine

Fig. 1. Lucas Combustion Chambers Are Shown Here Mounted in Position on the Rolls-Royce Derwent Jet Engine.

German flying bombs in June 1944. Figure 1 shows a later Rolls-Royce unit, the Derwent.

The Birmingham test house soon became too small for the volume of work, and in 1941, the Ministry of Aircraft Production decided to remove the organization to the vicinity of Burnley, Lancashire, as a protective measure against possible destruction by enemy action. A new building with improved and extended facilities was designed, and a second compressor was obtained to double the air capacity available at Birmingham. This new plant has a compressor house and an electrical substation, a well-equipped mechanical workshop, a sheet-metal workshop for the fabrication of prototype equipment, combustion test laboratories, and a number of fully instrumented test beds. The majority of this equipment is arranged to take air from the main compressors, the necessary pipe lines being provided to enable both compressors to operate in parallel on a single test bed, as is necessary in the case of the larger units now being developed. The majority of the test beds is arranged for the use of kerosene or similar fuels which, it is believed, will be the fuel of the future for gas turbines not only on account of the reduced fire risk but because of the increased calorific value for a given volume.

Special Test Beds

One test bed, however, is available in which provision has been made for the use of gasoline or other volatile fuel. All the controls are remote in a separate chamber, the test bed itself being isolated in a suitable fireproof enclosure.

In addition, a further bed is supplied with air from a Merlin blower driven by a 500-hp electric motor through suitable speed-increasing gear. This bed is used primarily for fundamental research work on combustion and carbon-formation problems.

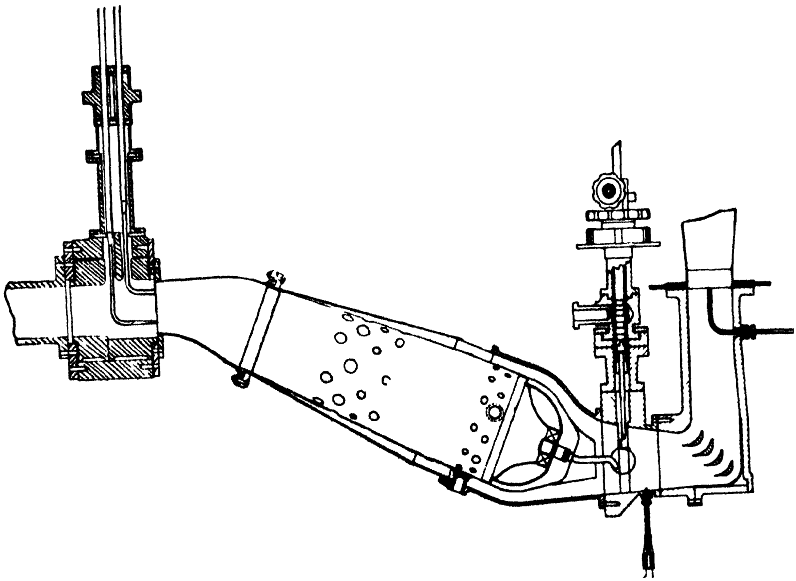
In studying conditions at lighting-up and starting, a simple atmospheric rig is supplied with air from an ordinary centrifugal blower. The combustion chamber discharges directly to atmosphere, so that the interior is visible, enabling the process of lighting-up to be studied. In addition, there is a chemical

laboratory equipped for complete gas analysis and the testing of fuels, and a physics laboratory dealing with the development of optical and other physical methods for air-flow tests, spectroscopy, and pyrometry.

As an indication of the rapid development of gas turbine design, it may here be mentioned that the existing facilities, complete though they are, will not suffice for testing the components of some of the units of more recent design in which much higher compression ratios are envisaged. To this end, it was decided to install two new compressors, each delivering six pounds of air per second at pressures up to 150 psi (gauge). These compressors, each driven by a 2250-hp electric motor,

Courtesy of Flight *Magazine*

Fig. 2. Component Parts and Assembly of Rolls-Royce Derwent Combustion Chamber.



Courtesy of Flight Magazine

Fig. 3 General Arrangement of Testing Rig with Cascade and Exhaust Traversing Gear.

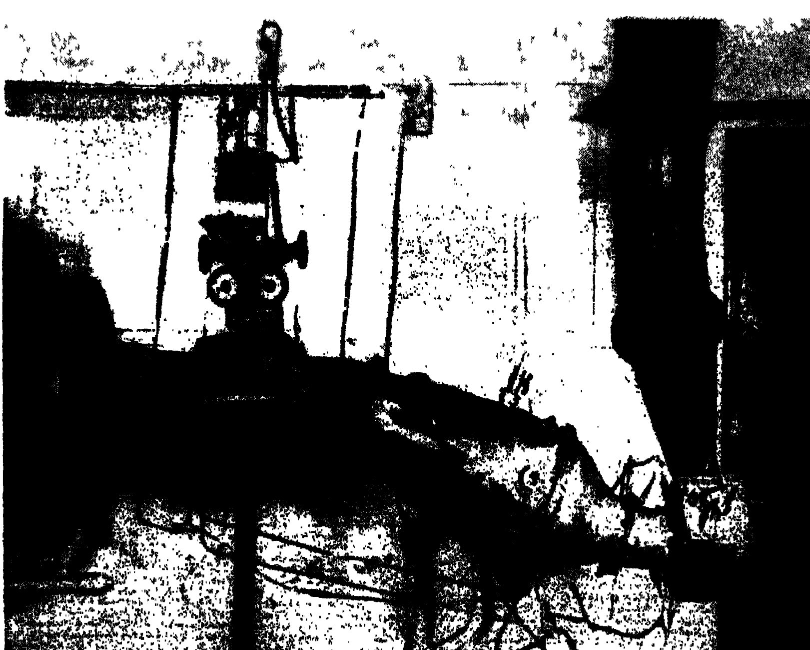


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Fig. 4. Rolls-Royce Nene Chamber on Test.

will make the laboratory capable of investigating combustion conditions for rather large gas turbine projects.

Figure 2 shows the component parts of a Lucas combustion chamber while Figs. 3 and 4 show a combustion chamber on test.

TABLE I

TEST PERFORMANCE OF TYPICAL CHAMBERS[a]

	No of Chambers	Fuel Consumption Gal Per Hr	Air Mass Flow Lb Per Sec	Air-fuel Ratio
Experimental	6	17 2	2.3	60:1
Derwent V	9	57	7	55.1
Ghost	10	67	8.8	59:1
Nene	9	81 5	10	55.1

[a] The figures of fuel and air flow relate to a single chamber.

The Fuel System

As the work on the jet-engine combustion system and equipment was proceeding, Joseph Lucas, Ltd., simultaneously was developing fuel pumps, burners, and regulating components for the control of the engine. This work was done in a specially equipped factory at Birmingham.

To meet the wide ranges of fuel flow required by the gas turbine at varying air speeds and altitudes, a pump having a variable stroke and regulated by a servo system under barometric control was deemed desirable. Since no light-weight pump occupying small space was available to meet these requirements, a special pump was developed, and was employed on all Rolls-Royce jet engines.

On the power unit, the fuel system includes a variable-stroke pump with a built-in maximum speed governor and an automatic pressure-relief mechanism, a barometric control unit, a throttle valve, a unit embodying accumulator, a trip valve and high-pressure cock, a fuel manifold, and burners. Disposition of this equipment on the wheel case at the forward end of the Rolls-Royce Derwent engine is shown in Figures 5 and

6, and connections can be readily followed in the diagrammatic layout shown in Figure 7.

Fuel is fed to the engine-driven pump, which operates in conjunction with the altitude-conscious barometric control

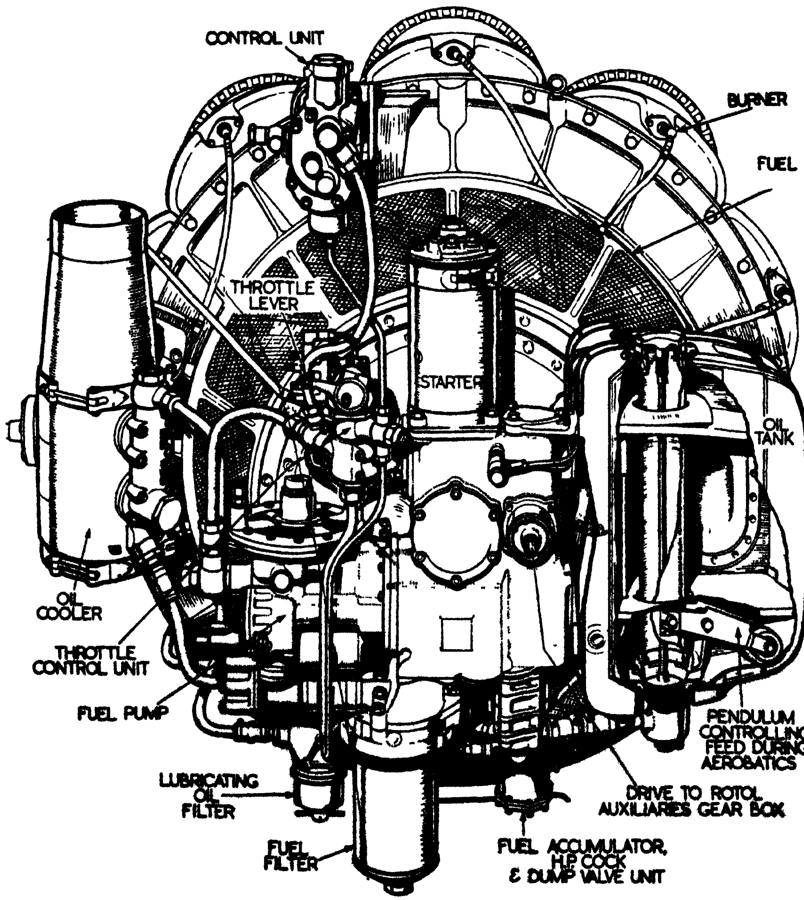

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Fig. 5. Fuel Injection and Control Components of Lucas Manufacture, Grouped on the Front of the Derwent Compressor.

unit and is subject to the overriding control of the hydraulic-type governor. The latter prevents the engine speed exceeding a predetermined maximum. From the pump, the fuel is delivered to the throttle valve, which is regulated by the pilot from

the cockpit, and thence to the accumulator unit, the ring manifold and, by individual flexible pipes, to the burners in the combustion chambers. The accumulator is required to build up a volume of fuel under adequate pressure for starting purposes.

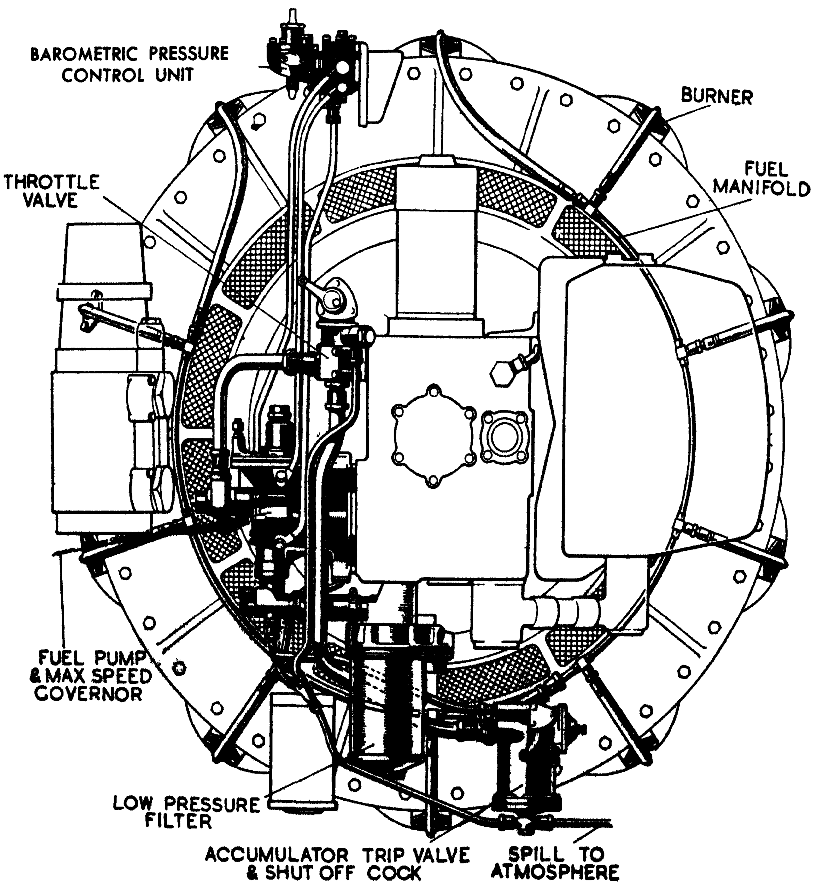

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Fig. 6. Installation of Fuel Equipment on Rolls-Royce Derwent Jet Engine.

When shutting down the engine, the high-pressure cock shuts off the flow of fuel to the burners. Drain connections from pump, throttle, and accumulator unit discharge to atmosphere. Gauges are provided to indicate pressure at the pump delivery,

manifold, and at the pump intake. In conjunction with the last, a pressure-sensitive switch operates a warning light in the event the fuel supply is interrupted by a clogged low-pressure filter.

The positive-displacement pump of Figure 8 has seven plungers which reciprocate in a rotor driven by the turbine shaft. Parallel with the rotor shaft is a bore in which the pressure-control piston operates. A hydraulic mechanism is in the pump to limit pump output and engine speed. The barometric control unit is also shown in Figure 8. This unit is used to assist the pump delivery pressure to oppose the spring load which is adjusted to give the desired regulation.

Throttle Valve

Located between the pump and the accumulator unit, the pilot's throttle valve is a tapered plunger valve in which the cross-sectional area of the flow space between plunger and seating is varied by axial movement of the plunger. Operation is positive because a pinion on the actuation lever spindle engages a rack cut on the plunger. When the throttle is closed, the plunger is fully seated but the engine is not shut down. A supply of fuel to maintain the engine running in the idling condition is by-passed from the inlet to the outlet by way of a spring-loaded valve opening automatically under fuel pressure. An adjustment screw limits the lift of this valve to determine the flow for slow running.

To shut down the engine, a high-pressure cock is provided to stop the fuel flow to the burners. This cock is embodied in the accumulator unit. At starting and while running, the cock remains in the fully opened position. When starting the engine, the cranking speed is insufficient to produce a delivery of fuel yielding a thoroughly atomized and ignitable spray at the burners. A temporary increase in the flow is, therefore, provided for starting purposes by the hydraulic accumulator. On switching on the starter motor, the engine is rotated at about 1200 rpm, giving a pump speed of about 200 rpm, and a fuel delivery at the rate of approximately 30 gal per hr. This discharge forces the accumulator piston downward against the

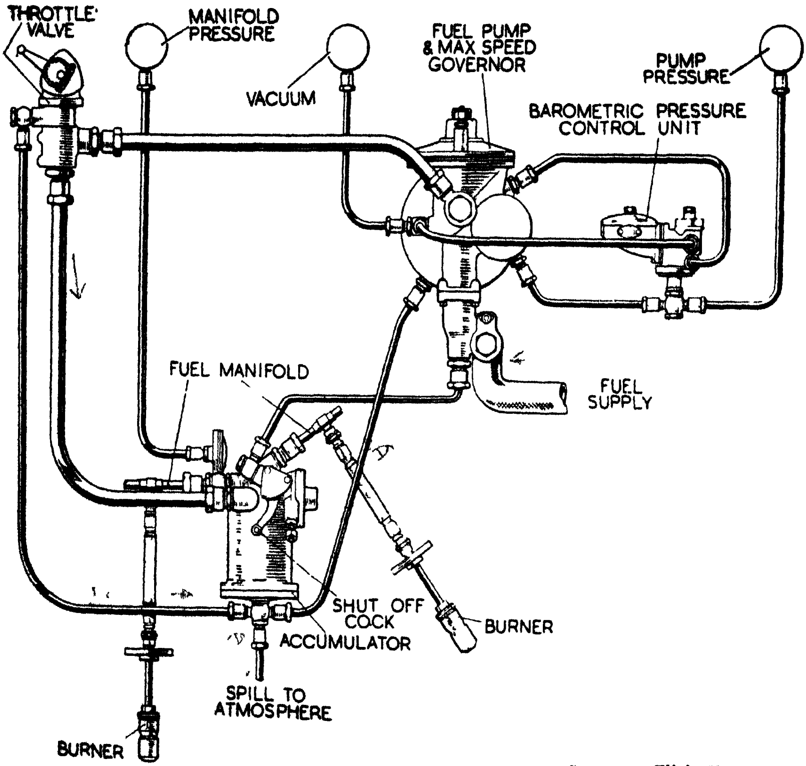

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Fig. 7. Simplified Diagram of Equipment Showing Connections.

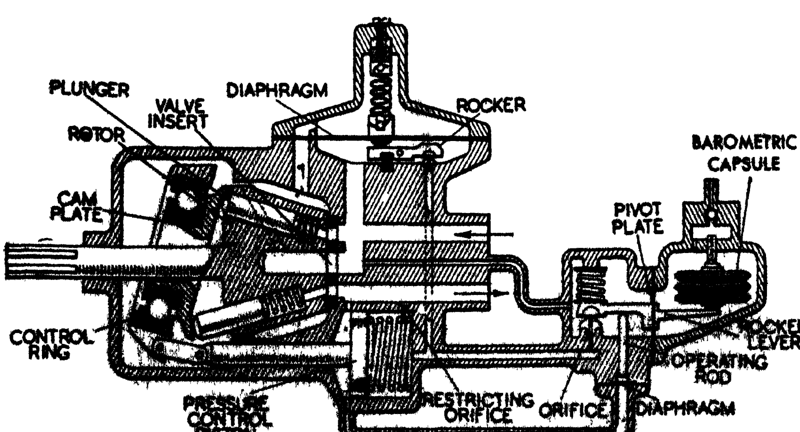

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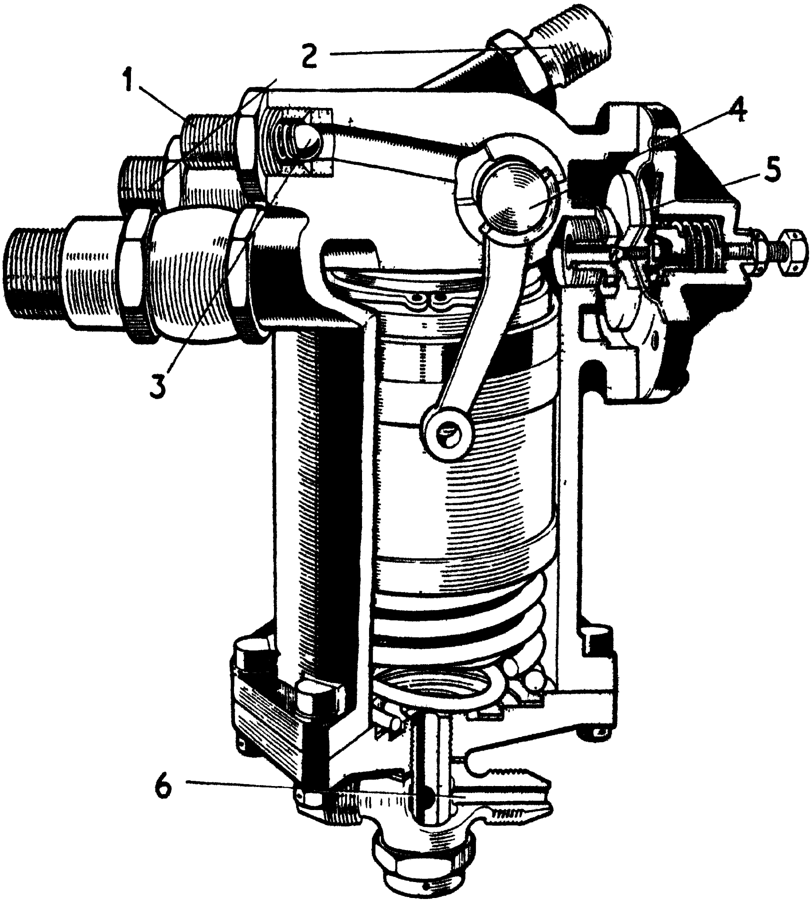
Fig. 8. Diagram of Pump and Barometric Pressure Control Unit Showing Interconnections.



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1—Connection to Pump Inlet
2—Connections to Manifold
3—Relief Valve
4—High-pressure Cock
5—Trip Valve
6—Spill to Atmosphere

Fig. 9. Accumulator, Trip Valve, and High-pressure Cock.

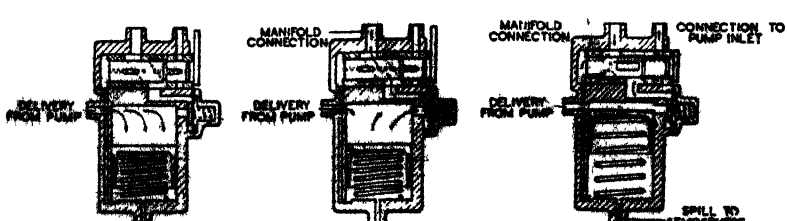

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Fig. 10. Flow Diagrams of Accumulator, Trip Valve, and High-pressure Cock.

loading springs, and when the accumulator is fully charged, the pressure lifts the trip valve against its spring. Immediately the trip valve leaves its seating, the pressure has effect on the full area of the supporting diaphragm, and the valve remains open until pressure has fallen to a very low value. Thus, there is a sharp discharge of fuel at a high rate of flow through the open high-pressure cock, the fuel manifold, and the burners. When ignition has occurred, the engine accelerates rapidly, and combustion is continued. Subsequently, the normal flow is across the accumulator and through the high-pressure cock (Fig. 9).

When the engine is shut down by closing the pressure cock, the accumulator discharges through the trip valve and along a slot in the rotor of the high-pressure cock to a ball-type relief valve, and is returned to the suction side of the pump. Simultaneously, the fuel in the manifold pipe is relieved through a hole drilled in the cock rotor, and then through a hole in the body to the underside of the accumulator piston, and is then spilled to atmosphere. As the accumulator discharges, the springs return the piston to its original position, the trip valve closes, and the unit is restored to the condition necessary for a further start. The flow through the accumulator unit under various conditions is shown in Figure 10.

Fuel Burners

The burner, mounted in the end of the combustion chamber and spraying downstream in relation to the air flow, is of the fixed-orifice, open type (Fig. 11). Fuel is passed through a tubular gauze strainer (which is supported by a spring so that if it becomes completely clogged, it will lift and be by-passed) to the interior of the burner body, in the end of which is mounted the atomizer. From the annular space between burner bore and atomizer, the fuel enters the conical atomizer chamber by three tangentially arranged holes, producing a rapid swirl. At the apex of the cone is a precisely calibrated orifice through which the fuel is projected in the form of a thin film which breaks up into a finely atomized spray. The atomizer is secured to the burner by a flanged sleeve drawn up by the air

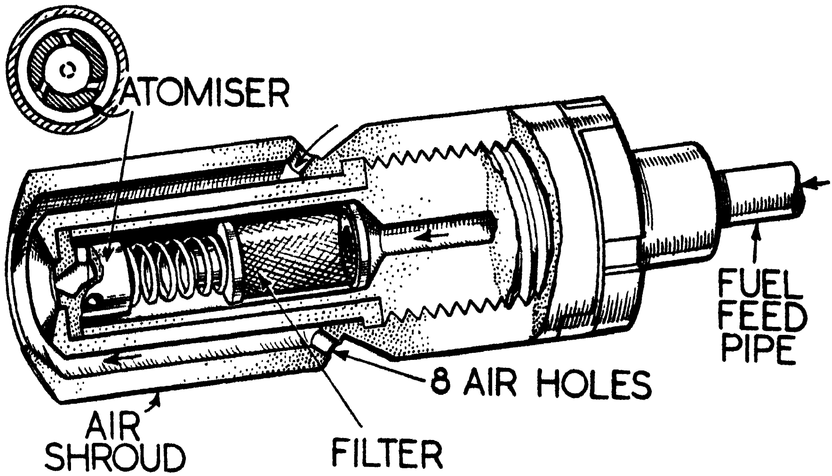

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Fig. 11. Fixed-orifice Burner with Sectioned Detail of Swirl Chamber.

shroud. The function of the shroud is to pass a controlled flow of air to the face of the atomizer to prevent the formation and build-up of carbon.

Starting

For starting the engine, a number of operations must be performed in a timed sequence. The problem is less complex for a turbine unit than for an equivalent piston engine, but to simplify the routine, an electrically operated system has been devised to carry through the starting cycle automatically on depression of a starter button.

The starting routine is as follows:

1. Close the master switch to energize the main supply circuit.

2. The check throttle is closed, and both low- and high-pressure fuel cocks are open.

3. Close the safety switch to energize control circuit, and start the supply pump in tank.

4. Depress the starter button for one second.

When the button is operated, the electric motor of the time switch is run to wind up the spring. Release of the button cuts out the motor, and the stored energy of the spring, operated

through a ratchet wheel, begins to turn the switch selector cams. First movement of the cams closes a pair of contacts to energize a magnetic relay, which in turn completes the circuits for the starter motor and the booster coils for the igniter plugs. The starter begins to rotate slowly and to take up play in the engaging clutch. Simultaneously, a second pair of contacts

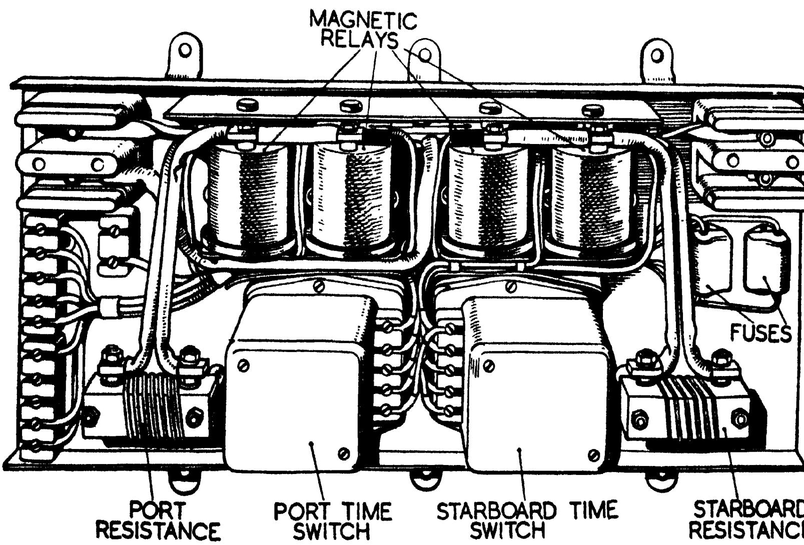

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Fig. 12. Starting Panel for Twin-engine Installation.

open, so that the push-button circuit is broken to prevent inadvertent operation before the cycle is completed.

Two or three seconds after the release of the button, a third pair of contacts is closed and another magnetic relay short-circuits the starting resistance and applies full voltage to the starter. After a total delay of 30 sec from the commencement of the cycle, the first and third contacts are opened and the second contacts closed to restore conditions for a repeat start. A sketch of the starting panel for a twin-engine installation is given in Figure 12. Usually, only two igniter plugs are employed, since once ignition has been initiated, the flame immediately flashes to other combustion chambers through the interconnecting tubes.

Section C—German

Chapter XXIII

Junkers Jumo 004 Aircraft Gas Turbine Jet Power Plant

Germany's jet propelled plane, the Messerschmitt Me-262, was the most successful enemy jet fighter encountered by the Allies in World War II. This airplane was powered by the Junkers Jumo 004 axial-flow gas turbine, a jet power plant which in design and manufacture reveals clearly how Germany's aircraft construction program was hampered by lack of skilled labor, materials, and adequate plant facilities.

Investigation reveals that most of these units had a service life of about 10 hr, but great progress was made in overcoming material difficulties. Immediately following enemy capitulation, it was found that Germany had developed a new alloy having excellent heat-resistant qualities, which made it possible to get up to 150 hr service in actual flight tests, and up to 500 hr on the test stand.

A large unit, the 004 is 152 in. long from the intake to the tip of the exhaust; is 30 in. in diameter at the skin around the six combustion chambers; and has a maximum diameter of the cowling reaching 34 in.

The circular nose cowling is double skinned, the two surfaces being welded together near the leading edge and held in position by riveted channel-shaped brackets. Diameter at the intake end is 20 in., the outer skin increasing to 31½ and the inner skin to 21½ in. Inside the cowling is an annular gasoline

tank which is divided into two sections, the upper being of ¾-gal capacity feeding fuel to the starting engine, the lower, of 3¾-gal capacity feeding starting fuel to the combustion chambers.

The nose cowling attaches by eight screws, in captured nuts, to the annular-shaped combination oil tank and cooler. Of three-gallon capacity, this tank has a baffle close to the inner surface so that, as warm oil is fed in from the top, it is cooled while it flows around to the bottom of annulus and the tank proper.

The oil tank, in turn, is attached by 23 bolts on a flange to the aluminum-alloy intake casting. This unit comprises the outer ring, with flanges on both front and rear faces, four hollow streamlined spokes, and the inner ring.

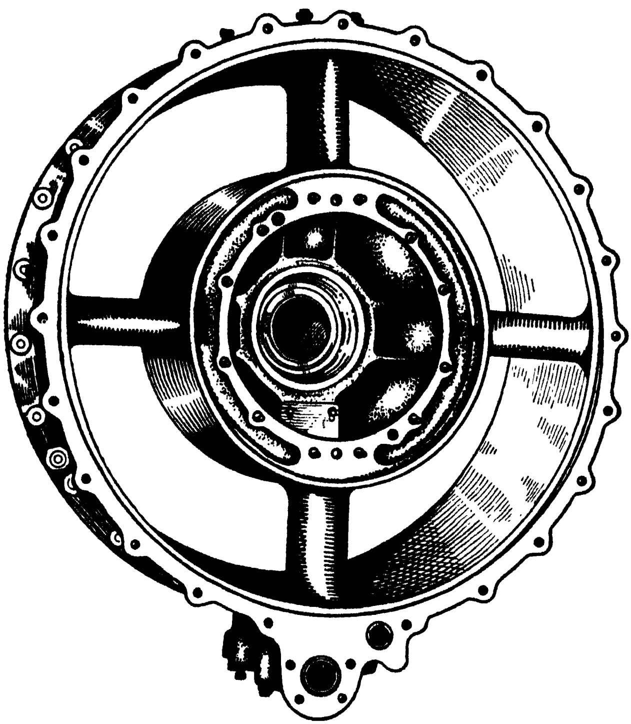
Inside the nose cowling, there is a fairing, increasing in size to 12 in. at the intake casting (Fig. 1), leaving approximately 200 sq in. of intake area. This fairing houses the starting engine, which is a two-cylinder, two-cycle, horizontally opposed gasoline engine developing 10 hp at 6000 rpm. The starting engine has its own electric starting motor. For use in emergencies, there extends out to the front of the shaft a cable starter similar to those found on outboard boat engines. The engine is 12½ in. long, 10 in. wide, 8¼ in. high, and weighs 36 lb.

The starter engine is bolted to six studs in the bevel gear casting, which contains gears to drive the accessories. Each of these gears is carried by ball and roller bearings, with the drive shafts fitting into internally splined stub shafts on the bevels. There are two drive shafts extending through two of the hollow fairings of the intake casting, one shaft going up to the accessory case (mounted atop the intake casting), the other extending down to the main oil pumps (set inside the lower part of the intake casting).

The bevel gear casting, also of aluminum alloy, is bolted to 12 studs set in a flange in the front face of the intake casting.

The rear side of the intake casting's inner ring is cup-shaped and houses the front compressor bearing. This unit is comprised of three thrust races each with 15 bearings. These

bearings are mounted in steel liners set in a light hemispheric-shaped housing which is kept in contact with the female portion of the intake housing by the pressure of 10 springs held in place by a plate bolted to the intake casting. The outer bearing races are mounted in separate sleeves which fit on the compressor shaft.



Courtesy of Aviation *Magazine*

Fig. 1. Front of Junkers Jumo 004 Intake Casting, with Oil Lines at Bottom.

This design not only allows for preloading the bearings during assembly to insure even distribution of thrust, but the bearing assembly can be left intact during disassembly simply by withdrawing the compressor shaft from the inner sleeve.

Next in the fore-to-aft sequence of location is the aluminum-alloy stator casting, which is built in top and bottom halves held together longitudinally by eleven $3/8$ in. bolts

through flanges on each side, with attachment to the intake casting by twenty-four 3/8 in. bolts through a heavy flange. Running the entire length of the bottom half of the casting are three 0.7 in. diam passages, one serving as part of the oil line leading to the rear compressor and turbine bearings, one connecting oil sumps (which are located in both intake and main castings), and one serving as part of the oil return line from a scavenge pump set in the rear turbine bearing housing.

Aft of the fourth compression stage in both halves of the stator casting is a slot, inside of which is a ring with a wedge-shaped leading edge pointing upstream and set to leave a 0.08 in. opening to bleed off air for part of the cooling system.

Like the stator casting, the stator rings, which consist of inner and outer shroud rings and stator blades, are built as subassemblies and then are bolted in place and locked by small tabs.

Considerable variation, both in materials used and methods of construction, was found in this section. In early production units, for example, the inlet guide vanes and first two rows of stator blades were of stamped aluminum with airfoil profiles; and in assembly, ends of the blades had been pushed through slots in the shroud rings and brazed in place. In other early engines, the third stator row varied both in material and method of attachment. In some cases, this row would be of aluminum, but without airfoil; in others, the third stator row would be of steel with the ends turned to form flanges, which were spot-welded to the shroud rings. The remainder were stamped zinc-coated, sheet steel.

One late-production engine examined showed a combination of all the variations, with inlet guide vanes and first two rows of stator blades of stamped aluminum, and the remainder steel, indicating that the Germans may have been substituting steel exclusively for aluminum. Apparently, all the steel blades had been enameled, but this protective coating on the last row, where temperatures reached approximately 380° C, appeared to have been burned off.

Methods of attaching blades to shroud rings also varied. On the inlet guide vanes and first two rows, the ends of the

blades had been pushed through slots in the shroud rings and brazed in place; the third, sixth, and seventh rows had a weld completely around the blade end; the fourth, fifth, and eighth rows of blade ends had been formed into split clips which were spot-welded to the shroud rings.

The outer shroud rings are channel-shaped with an angle bracket riveted to each end, this bracket in turn being bolted to a stud set in the casing just inside the mating flange. Inner shroud rings are flanged along the leading edge, with the exception of the seventh row, which is channel-shaped.

Except for the inlet guide vanes and the last row of stator blades, which act as straighteners, stator blades are arranged as impulse blading—they are set at nearly zero stagger and simply serve as guides to direct the air flow into the rotor blades.

The compressor rotor is made up of eight aluminum disks held together by 12 bolts, each through shoulders approximately at mid-diameter, with the entire unit being pulled together by a 38.75 in. long, 0.705 in. diam tie rod which has been estimated to have a stress of some 40,000 psi, and with a force to pull the assembly together figured at about 16,000 psi (Fig. 2).

Diameters of the disks increase from the low- to high-pressure ends as follows: Stage 1, 13.86 in.; Stage 2, 14.68 in.; Stage 3, 15.61 in.; Stage 4, 16.44 in.; Stage 5, 17.18 in.; Stage 6, 17.85 in.; Stage 7, 18.24 in.; and Stage 8, 18.34 in.

To carry the compressor bearings there is attached to each end disk a steel shaft with an integral disk carrying a round-faced washer. This shaft goes through the disk and is tightened by a nut so that the face of this washer (rounded to facilitate alignment) bears against the disk face. The flange on the rear shaft has six slots around its outer edge, into which fit projections on the rear disk. Thus, torque is transmitted from the turbine to the rear compressor disk, and from there on to the other disks by the bolts previously noted as fastening the disks together, the torque being transmitted to the compressor unit around the faces, rather than through a central shaft.

Compressor rotor blades, of which there are 27 in the first

two stages and 38 in the others, are all stamped aluminum with machined roots fitting into pyramid-shaped slots in the rotor disk. Through the aft face of each blade root, directly

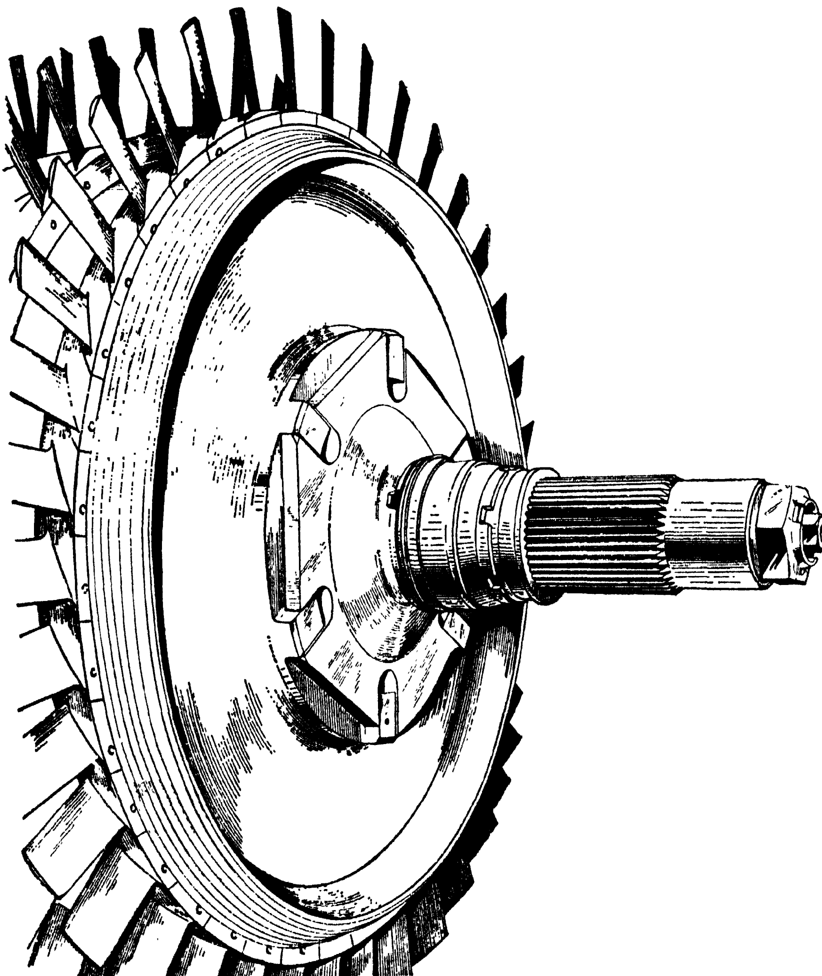


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Fig. 2. Compressor Disk and Rotor.

under the blade trailing edge, is a small screw set longitudinally and extending into the disk.

Tip stagger of the blades is about the same through the first six stages of compression, but increases in the last two.

Chord of the blades decreases through the eight stages as follows: 1.95, 1.94, 1.34, 1.33, 1.30, 1.30, 1.24, and 1.21 in.

Blade profiles in the first two stages are very similar (possibly even designed to the same section), but the third stage has a thicker section. Stages 4, 5, and 6 have thinner sections (here, too, possibly the same), with about the same chord as Stage 3, and the last two stages, though set at greater pitch and having slightly narrower chord, have generally similar camber and profiles.

Clearances between the rotor blades and the stator casting are 0.103 in. over the first three stages and 0.04 over the remaining five. Axial clearances between rotor disks and inner stator shroud rings range from 0.1 to 0.15 in., and axial clearances at the roots between rotor and stator blades are 0.5 and 0.6 in.

Backbone of the 004 is a complex aluminum casting which, in addition to providing the three engine attaching points, supports the compressor casing (through 25 bolts), the entire combustion chamber assembly, the turbine nozzle, the aft compressor bearing, the two turbine bearings, and, through the combustion chamber casing, the entire exhaust system. Moreover, in the base of each of the six ribs supporting the combustion chambers, there are cored passages, five of which carry cooling air, one carrying lubricating oil. And, although the air passage area remains constant between the compressor and combustion chambers, the main casting changes the shape from annular to circular at the entrance to the chambers.

In the front of the casting, at the tip of the last stator row, is an 18⅜ in. diam ring with a serrated inner surface fitting closely to serrations on the aft face of the compressor disk. Air bleeding through the serrations is carried aft through cored holes in the casting to cool the front face of the turbine disk and, on hollow-bladed turbines, to cool the blades themselves.

Just outside and in back of this ring are the fairings which divide the air and direct it into the individual combustion chambers. These fairings, in turn, are surrounded by a ring having an outside diameter of 28 in. and containing 25 bolt holes for attaching the compressor casing. In addition to the

bolt holes, there are 18 openings, six of which carry the air bled off from the compressor to the rear for exhaust system cooling, and 12 smaller openings which take cooling air around the combustion chambers.

Around the outside of this ring, and extending back to a heavy flange to which the combustion chamber casing is bolted, are 12 raised longitudinal ridges arranged in pairs. These ridges have machined faces, the latter containing four bolt holes and two aligning pins to serve as the forward-engine pickup points. Since there are six such pickup points, the engine was designed for a wide variety of mountings. In the case of the Me-262, plates with collared nuts were fastened to the two bolts located on either side of the topmost unit.

Over-all length of the main casting is 37¼ in., with the previously mentioned ribs tapering down from the aft face of the ring structure to the central longitudinal member, which has an 8¾ in. diam at the aft end.

The aft compressor bearing, having 16 rollers, is set in the front of the main casting inside the serrated ring, the housing being attached to the casting by 14 bolts.

The turbine thrust bearing is set inside the main casting, with the center line of the balls 24⅜ in. back of the front edge of the serrated ring. The main turbine roller bearing is bolted into the rear end.

Each of the six combustion chambers is built up of three major components having a combined weight of 19 lb. First, there is a mild-steel outer casing 20⅝ in. long, 5¾ in. in diameter at the entering end, flaring out to 8⅝ in. The front end has a collar with a rubber sealing ring which is pushed up against the aft face of the main casting to take care of air leakage and to compensate for the difference in casting and combustion chamber expansion.

Fitting inside the front end of this casing is the flame tube, which has two main components—the entry section and stub-pipe assembly. The fore part of the entry section flares out somewhat as does the outer casing, and at the front end has a six-blade swirler. This unit is made of 22-gauge mild steel with a black enamel coating. The stud-pipe assembly is made up of

10 flame chutes welded to a ring (which is, in turn, welded by brackets to the rear end of the flame tube) and to a four-inch dished baffle plate at the rear. In order to help direct air into the chutes, ½ in. circular baffle plates are riveted to the forward ring. Material of this unit is mild steel with an aluminized finish.

The third major component of the combustion chamber is an 11 in. long, 20-gauge aluminized steel liner having a corrugated outer skin which permits cooling air to flow inside the

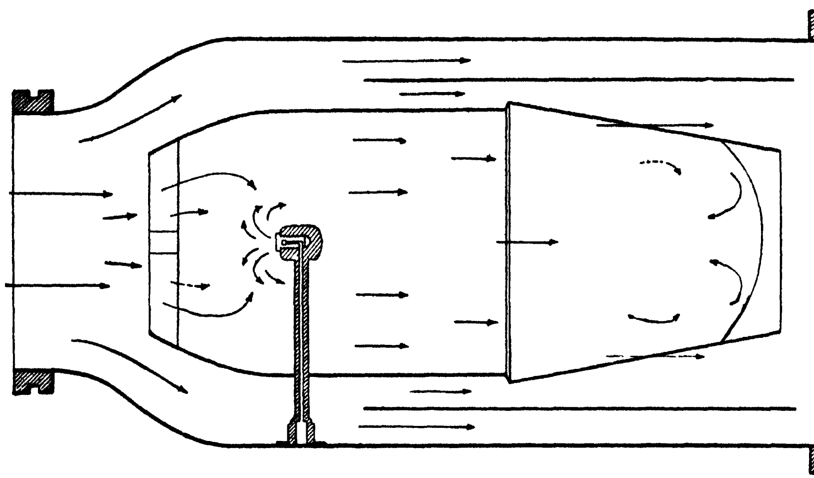

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Fig. 3. Exploded View of Combustion Chamber Showing Main Components.

outer casing. This liner fits into the aft end of the casing. The aft ends of the combustion chambers are bolted around flanges to a ring of six rings which fits over the rear end of the main casting.

Ignition interconnectors between chambers are of but 15⁄32 in. diam, and starting plugs are provided in three of the six chambers. These elements, as are the fuel plugs, are enclosed in streamlined fairings.

Surrounding the combustion chambers is a 16-gauge mild-steel double-skinned casing having flanges welded at both ends —that at the front end attaching by studs to the main casting and that at the rear attaching to the turbine inlet-duct outer

flange, the nozzle-ring-assembly flange, and the exhaust casing flange. Besides the bolt holes in the front flange, there are 24 of similar size. Twelve bolt holes lead to six ducts of 22-gauge steel which carry the air bled from the fourth compressor stage through the combustion chamber casing, and 12 direct air around the combustion chambers. These ducts also help stiffen the skin, since it takes the weight of the entire exhaust system (Fig. 3).

Six large hand holes are cut in the casing immediately behind the flange. These holes give access for making minor adjustments to burners and the three ignition plugs.

A little more than halfway aft around the combustion chamber casing is a heavy collar comprised of two channel-shaped members, and inside the casing at this ring are six tie rods which connect the combustion chamber casing to the main casting. Any one of these six units can serve as the aft-engine pickup point; in the case of the Me-262, the top unit performs this function.

Ducting from the combustion chambers to the turbine nozzle changes the air passage from the six circles to annular shape. Attached to the combustion chambers by bolts, this 19-gauge aluminized mild-steel unit is made in two parts, the rear of which is welded to a heavy flange. Studded to this flange from the inner shroud ring of the turbine-nozzle assembly are two mild-steel diaphragm plates. These, in turn, are studded to the rear end of the main casting, and so support the inlet ducting and turbine nozzle ring. On the rear of the outer-turbine inlet ducting a light flange mates with a flange on the rear of the combustion chamber casing. Thus, the turbine inlet ducting, to which the combustion chambers are attached, is supported partly by the main casting, partly by the diaphragms, and partly by the skin.

Maintenance crews take a beating as the result of the final design, for it is a major operation to get at the combustion chambers. First, the variable-area nozzle operating shaft must be removed so that the complete exhaust-system assembly can be taken off. Following this, unless special equipment is available, the engine must be placed upright on the turbine disk

and burner pipes and ignition leads disconnected from the combustion chambers. Then, the compressor casing-main casting joint can be broken and the whole front end of the engine lifted off. Next, the rear compressor bearing assembly, torque tube, and locking ring can be removed. The main casting assembly can be taken out after the nut on the front end of the turbine shaft has been unscrewed. The rear diaphragm plates can then be removed and the turbine inlet ducting and combustion-chamber assembly lifted off. Finally, the front diaphragm plate, the turbine inlet ducting, and the combustion-chamber assembly are lifted out of the casing. At this point, as one sweating engineer who did the job declared, "Now, Bub, y'can take out the individual combustion chambers."

An unusual feature of the 004's design is the use of hollow turbine nozzle blades through which cooling air is fed from the compressor through the main casting and supporting diaphragm plates. The two-part outer nozzle shroud ring is made of mild steel and both parts are welded to a ring that is joggled and flanged to mate with flanges through 36 bolts on the inlet ducting and the aft flange of the combustion chamber casing. In addition to the bolt holes, the flange has 36 sets of three holes for cooling air passage (Fig. 4).

The 35 nozzles are made of 0.045 in. thick austenitic sheet steel bent to shape around a $\frac{1}{16}$ in. radius to form the leading edge. Between the sheets at the trailing edge are spot-welded four one-inch wedge-shaped spacers, that taper from $\frac{1}{8}$ to 0.020 in., leaving a 0.020 in. gap down the trailing edge through which the cooling air escapes.

In assembly, the blade tips are closed, pushed through slots welded to the outer shroud ring, and the roots are pushed through slots in the inner shroud ring and spot welded in place on the inner surface of the ring.

To this ring, in turn, is welded a heavy, mild-steel flange and a second flanged ring. The two flanges pick up with the diaphragm plates which support the assembly from the rear of the main casting.

Two types of 61-blade turbines are used. Originally both

blades and disks were solid; later, hollow blades and lighter disks were introduced at a saving of approximately 40 lb.

The solid disks were of hardened chrome steel which took stresses of about 15 tons at maximum rpm. Cooling was effected

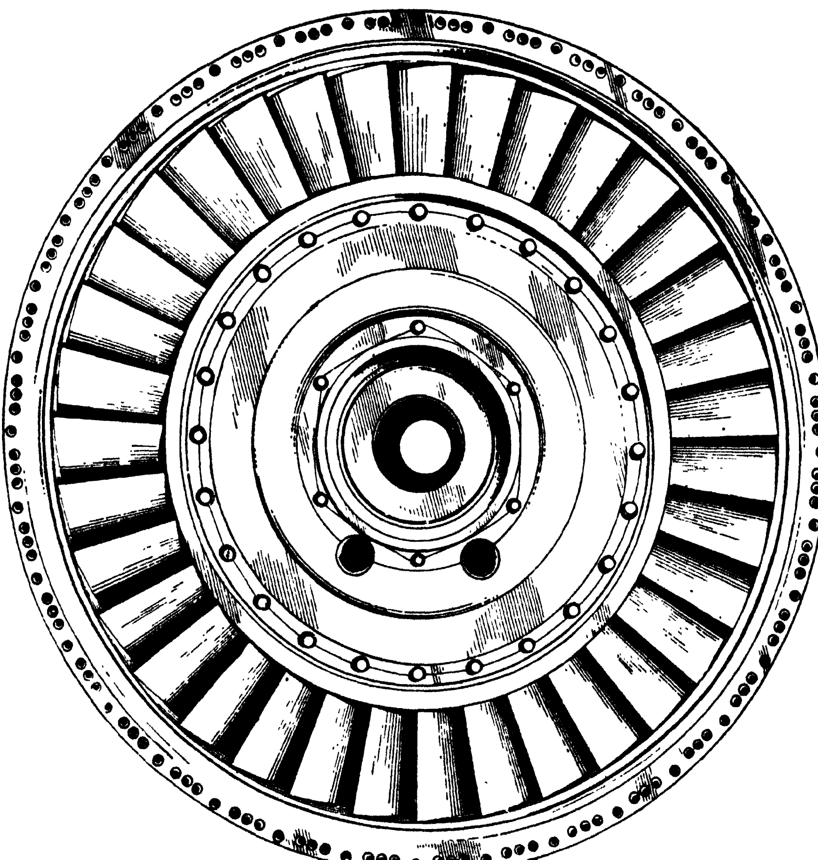

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Fig. 4. Front View of Turbine Nozzle Assembly Showing Rear Turbine Bearing and Diaphragm Plates in Place.

by spilling air that was bled back through the main casting against the disk face and then up over the blade roots and out between the blades.

The 12¼ oz solid blades are forged from an austenitic steel containing 30 per cent nickel, 14 per cent chrome, 1.7 per cent

titanium, and 0.12 per cent carbon, corresponding closely to *Tinidur,* a Krupp alloy known before World War II. The blades are attached by three machined lugs drilled to take two 11 mm rivets each. Maximum centrifugal blade stresses have been estimated at 18,000 psi, and gas bending stresses at 2000 to 4000 psi. Study of the solid blades indicates that the roots did not attain a temperature much higher than 450° C, owing to the cooling air flow up from the disk, but near the center, the temperature of the blades reached approximately 750° C. This applies to service models, not those previously mentioned as having given the longer flight and test-stand life.

Disks for hollow-blade turbines are of lighter material than for the solid types, and have attached, across the front face, a thin sheet flared out near the center. This picks up the cooling air and, through ridges on the disk, whirls it out toward the blade roots where the air goes through two small holes drilled in the disk rim, up through the blade, and out the tip.

Made of the same material as the solid blades, the hollow type are formed by deep-drawing a disk through 15 operations. In assembling the turbine, the blade roots are fitted over grooved subs on the disk rim. Two small holes on each side take locating pins to hold the blades in place during assembly, but these pins take no stresses.

Having placed a silver-base flux in the grooves, the entire unit is put in an oven at 600 to 800° C, warmed for 20 min, heated to about 1050° C for 40 min., and then cooled in still air at room temperature before being hardened in a gas or air oven.

Later production units have two rivets in the blade trailing edges near the tips, a modification made necessary by cracking caused by vibration.

The turbine is attached by six studs to a short shaft carried on two bearings housed in the main casting. The front bearing is a single-race ball thrust, the rear, a single-race roller type, and both are cooled by oil only. Connection of the turbine and compressor is through a heavy internally splined coupling.

The exhaust cone is made of aluminized mild steel, and consists of two major components—outer and inner fairings.

The outer fairing is double-skinned, with cooling air bled from the compressor flowing between the skins to within 15¾ in. of the exit where the inner skin ends. Outside, the other skin is flared at the leading edge to scoop in cooling air. It is attached by spot-welded corrugations.

Attached to the outer fairing by six faired struts is the inner fairing, tapering from 19½ in. at the turbine end to 9¾ in. This unit houses a rack gear driven by a shaft entering through one of the struts which moves a *bullet* extending from its aft end (Fig. 5). Actuating this bullet over its maximum travel of approximately 7⅜ in. varies the exit area between 20 and 25 per cent. The bullet is set in retracted position for starting to give greater area and help prevent overheating; then the bullet is moved aft to decrease the area and give greater velocity for take-off and flying. The movement is accomplished by a gear-type servo motor placed near the accessory housing and connected by a long torque tube to gears placed on the exhaust housing over one of the struts leading into the previously mentioned rack gear.

Courtesy of Aviation *Magazine*

Fig. 5. Top View of "Bullet" which Moves Fore and Aft in Exhaust Cone to Give Variable Area Exit.

Originally, the unit was intended to operate automatically over small ranges at extremely high speeds and altitudes to give maximum efficiency. However, on some engines examined, the necessary lines had been blanked-off. The two-position operation is obtained through a mechanical linkage with the throttle so that the bullet moves aft at between 7000 to 7500 rpm.

Since the necessary cooling system played a very important part in both the design and construction of the 004, this system is described briefly below. It consists of three major stages, as follows (Fig. 6):

1. Air bled off after the fourth compression stage.

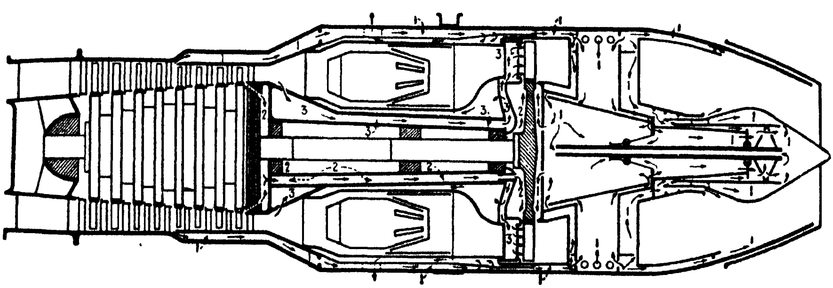

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Fig. 6. Schematic Diagram of Cooling System, which Takes Well Over 7 Per Cent of Total Air Intake.

2. Air taken off just after the last compression stage.
3. Air bled off between the compressor and combustion chambers.

In Stage 1, the air is picked up by the ring after the fourth compressor row and is directed into six cored passages in the stator casting. Then, at the combustion chamber casing, it is divided so that some of the air goes through six ducts in the combustion-chamber casing skin and some goes inside the casing and around the chambers themselves. The air going into the ducts continues aft and, through small holes in the flanges, enters between the double skin of the exhaust-cone outer fairing. Most of the air goes straight on aft to the end of the inner skin, but some is taken through the six struts connecting the inner fairing into that fairing to cool the rack gear and bullet.

In Stage 2, the air goes through the serrations between the compressor and the main casting, into two of the six cored passages in the casting back to the turbine. Here, on the original engines, the air was spilled against the face of the turbine disk and moved out to escape between the turbine blades. On engines with hollow blades, however, the air is ducted across the space between the two diaphragm plates supporting the turbine nozzles, then inside the sheet attached to the turbine disk, where it is picked up by ridges, forced up through the turbine blade roots, and out through the blade tips.

Stage 3 cooling air, bled off between the compressor and combustion chambers, is ducted through three passages in the main casting to the space between the turbine nozzle-supporting diaphragms, then up through the turbine nozzle vanes into the slipstream through the trailing edges of the vanes.

It is estimated that Stages 1 and 3 take approximately three per cent each of the total air movement, and that Stage 2 probably takes at least half as much. Thus, more than seven per cent of the available flow is taken off because of a lack of higher heat-resistant alloys. Additional performance penalties are evident in the fact that ducting is necessary, which complicates both the weight and production pictures.

Air is not the only cooling medium, because the lubricating system, too, is employed. In this system, two gear pumps circulate lubricating oil to the front compressor-drive bevel gears, and the accessory gears. Another pump supplies oil to lubricate and cool the rear compressor and both turbine bearings, the latter two being sprayed and splashed, respectively.

The two main pumps, mounted beneath the engine and driven from the bevel gears through a nose casting strut, deliver 190 gal per hr each. The two-part scavenge unit is built into the turbine bearing housing and is driven by a gear cut into the sleeve, which serves to return oil to the cooler. In level flight, one part of the unit, a 300 gal per hr pump, returns oil through one of the cored passages in the main casting, then, through a passage in the stator casting, to the pump in the bottom of the intake casting. In climbs, the other part of the scavenge unit, a 90 gal per hr gear pump, picks up the oil and

feeds it into a common return line to the air-oil separator. Oil is returned from the main pump to the separator by a 300 gal per hr pump driven by the same shaft as the delivery pumps.

Two types of fuel are used on the 004—gasoline for starting and J-2 brown coal *crud* for running. The gasoline is carried in the lower part of the annular tank set in the nose cowling, and is automatically cut off after ignition at about 3000 rpm. This gasoline is fed by an electrically driven pump delivering 90 gal per hr at 28 psi. Toward the end of World War II, it was found that centrifugal crude oil was also used as operating fuel.

The main single-stage electrically driven gear-type pump has a maximum delivery of 500 gal per hr at 1000 psi, at 3000 rpm.

Most interesting of the accessories is the all-speed governor, a 17 lb unit consisting basically of a centrifugal governor, oil pump, and spill and throttle valves. In operation, oil goes through a passage to the pilot piston, and is distributed to outer faces of either the spill or follow-up piston, depending on movement of the flyweights. Both the pistons move at the same time, adjusting the fuel spill to counteract changes in engine speed. The distance between the spill and follow-up pistons varies according to the flow of oil through the passages, so that the spill-piston action is a step-by-step operation controlled by the follow-up which returns to normal position after each step. A throttle valve is linked with the governor cam, so that when the throttle is advanced, the fuel flow increases,

General Data

Weight (without cowl)	1669 lb
Weight (with cowl)	1775 lb
Specific weight	0.85 lb
Thrust	1070 to 1980 lb
Pressure ratio	3 to 1
Fuel consumption	2720 to 2745 lb per hr
Specific fuel consumption	1.375 to 1.39
Maximum speed	8700 rpm
Idling speed	3080 rpm
Idling-speed fuel consumption	614 lb per hr
Length	152 in
Maximum diam	34 in
Frontal area	6.4 sq ft

COMPRESSOR DATA

Number of Stages	Number of Blades Stator	Number of Blades Rotor
Inlet Guide	32	
1	61	27
2	61	27
3	59	38
4	61	38
5	61	38
6	71	38
7	71	38
8	57	38

and response is immediate. The governor then takes over and adjusts the engine speed to a predetermined value set by the position of the cam.

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